Notice No. 1

Rules and Regulations for the Classification of Naval Ships, January 2016

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Issue date: June 2016

Amendments to	Effective date
Volume 1, Part 1 Chapter 1	1 July 2016
Volume 1, Part 1, Chapter 3, Section 4	1 July 2016
Volume 1, Part 3, Chapter 3, Section 2 & 3	1 July 2016
Volume 1, Part 3, Chapter 4, Section 7	1 July 2016
Volume 1, Part 5, Chapter 4, Section 6	1 July 2016
Volume 1, Part 6, Chapter 3, Section 4	1 July 2016
Volume 1, Part 6, Chapter 6, Section 1	1 July 2016
Volume 2, Part 4, Chapter 3, Section 4	1 July 2016
Volume 2, Part 4, Chapter 4, Section 5	1 July 2016
Volume 2, Part 7, Chapter 1, Section 5	1 July 2016
Volume 2, Part 7, Chapter 3, Section 5	1 July 2016
Volume 2, Part 8, Chapter 1, Section 15	1 July 2016
Volume 3, Part 1, Chapter 1, Section 2	1 July 2016

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Volume 1, Part 1, Chapter 1
General Regulations

■ *Section 8*
Limits of Liability

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8.5 Any dispute about LR services is subject to the exclusive jurisdiction of the English courts and will be governed by English law.

Volume 1, Part 1, Chapter 3
Periodical Survey Regulations

■ *Section 4*
Docking Surveys and In-water Surveys

4.3 In-water Surveys

4.3.1 The Committee will accept an In-water Survey between Special Surveys, as a Docking Survey, where suitable protection is applied to the underwater portion of the hull. If requested, the ***IWS** notation may be assigned on satisfactory completion of the survey, provided that the applicable requirements of the Rules are complied with.

4.3.2 The In-water Survey is to provide the information normally obtained from the Docking Survey, *see Vol 1, Pt 1, Ch 3, 4.2 Docking Surveys 4.2.2*.

4.3.3 The underwater part of the hull should be marked with search lines for reference purposes.

4.3.4 The In-water Survey is to be carried out at an agreed geographical location under the surveillance of a Surveyor to LR, with the ship in sheltered waters and with weak tidal streams and currents. The in-water visibility is to be good and the hull below the waterline is to be clean. The Surveyor is to be satisfied that the method of pictorial presentation is satisfactory by use of CCTV. There is to be good two-way communication between the Surveyor and the diver.

4.3.5 Prior to commencing the In-water Survey, the equipment and procedures for both observing and reporting the survey are to be agreed between the Owners, the Surveyor and the diving firm.

4.3.6 The In-water Survey is to be carried out by a qualified diver employed by ~~the~~ a firm approved by LR. In addition, for certain aspects of the In-water Survey, consideration may be given to the use of a remotely operated vehicle (ROV) operated by the LR approved firm.

4.3.7 If the In-water Survey reveals damage or deterioration that requires early attention, the Surveyor may, in consultation with the Owner, require that the ship be dry-docked in order that a fuller survey can be undertaken and the necessary work carried out.

4.3.8 Where a vessel has the ***IWS** notation, the condition of the high resistant paint is to be confirmed at each dry docking in order that the ***IWS** notation can be maintained.

Volume 1, Part 3, Chapter 3
Ship Control Systems

■ *Section 1*
General

~~**1.4 Materials**~~

~~1.4.1 The requirements for materials are contained in the *Rules for the Manufacture, Testing and Certification of Materials, January 2016* (hereinafter referred to as the Rules for Materials).~~

~~1.4.2 Rudder stocks, pintles, coupling flange bolts, keys and cast parts of rudders are assumed to be made of rolled, forged or cast carbon manganese steel in accordance with the *Rules for the Manufacture, Testing and Certification of Materials, January 2016*. Where other materials are proposed the scantlings will require to be specially considered on the basis of the Rules.~~

~~1.4.3 For rudder stocks, pintles, keys and bolts the minimum yield stress is not to be less than 200 N/mm². The following requirements are based on a material's yield stress of 235 N/mm². If material is used having a yield stress differing from 235 N/mm² the material factor is to be determined as follows:~~

$$\cancel{K_o = \left[\frac{\sigma_o}{235}\right]^m}$$

~~where~~

~~m = 0,75 for σ_o > 235 N/mm²~~

~~m = 1,0 for $\sigma_o \leq$ 235 N/mm²~~

~~σ_o is as defined in *Vol 1, Pt 3, Ch 3, 1.2 General 1.2.1*~~

~~1.4.4 In order to avoid excessive edge pressures in way of bearings, rudder stock deformations should be kept to a minimum. Where significant reductions in rudder stock diameter due to the application of steels with yield strengths exceeding 235 N/mm² are proposed, final acceptance may require the evaluation of the rudder stock deformations.~~

■ *Section 2*
Rudders

2.1 Application

2.1.1 This Section applies to ordinary profile rudders, and to some enhanced profile rudders with special arrangements for increasing the rudder force, as defined in *Table 3.2.2 Rudder profiles*.

2.2 Design considerations

2.2.1 Effective means are to be provided for supporting the weight of the rudder without excessive bearing pressure, e.g. by a rudder carrier attached to the upper part of the rudder stock. The hull structure in way of the rudder carrier is to be suitably strengthened.

2.2.2 Suitable arrangements are to be provided to prevent the rudder from lifting.

2.2.3 In rudder trunks which are open to the sea, a seal or stuffing box is to be fitted above the deepest load waterline, to prevent water from entering the steering gear compartment and the lubricant from being washed away from the rudder carrier. If the top of the rudder trunk is below the deepest load waterline, two separate stuffing boxes are to be provided. Rudder trunk

boundaries, where exposed to the sea, are to have a corrosion protection coating applied in accordance with the manufacturer's instructions.

2.2.4 Where the top of the rudder tube is significantly higher than the deepest load waterline a lesser arrangement of watertightness, such as 'O' rings may be accepted.

2.2.5 The watertight gland body may be formed by the top of the fabricated or cast rudder tube; the gland packing being retained against the top bearing or a check in the wall of the rudder tube and is compressed by a gland packet which may be of the flange type, screwed cap or other suitable arrangement.

2.2.6 Alternative arrangements utilising lip seals or 'O' rings either in isolation or in combination with one or other of the alternative seal arrangements will be the subject of special consideration.

2.3 Materials

2.3.1 The requirements for materials are contained in the *Rules for the Manufacture, Testing and Certification of Materials, January 2016*.

2.3.2 Stern frames, rudder horns, shaft brackets, rudder stocks, pintles, coupling bolts, keys, and other rudder members are to be made of rolled, forged or cast carbon-manganese steel in accordance with *Ch 3 Rolled Steel Plates, Strip, Sections and Bars*, *Ch 4 Steel Castings* and *Ch 5 Steel Forgings* of the *Rules for the Manufacture, Testing and Certification of Materials, January 2016*.

2.3.3 For rudder stocks, pintles, coupling bolts and keys the minimum yield stress is not to be less than 200 N/mm^2.

2.3.4 For all parts of the rudder, including rudder stocks, pintles, coupling bolts and keys having a specified minimum yield stress differing from 235 N/mm^2, the material factor, *k* is to be determined in accordance with *Table 3.2.1 Rudder material factor, k*:

Table 3.2.1 Rudder material factor, *k*

Specified minimum yield stress, σ_o (N/mm^2)	k
$\sigma_o > 235$	$k = \left(\frac{235}{\sigma_o}\right)^{0,75}$
$\sigma_o \leq 235$	$k = \frac{235}{\sigma_o}$
Note 1. The specified minimum yield stress is not to be taken as greater than 70 per cent of the ultimate tensile strength. Note 2.The specified minimum yield stress is not to be taken as greater than 450 N/mm^2.	

2.4 Welding and design details

2.4.1 Slot-welding is to be limited as far as possible. Slot welding is not to be used in areas subject to large in-plane stresses transverse to the slots or in way of cut-outs of semi-spade rudders. Continuous butt welding with backing may be accepted in lieu of slot welds. When continuous butt welding is applied, the root gap is to be between 6-10 mm. The bevel angle is to be at least 15°.

2.4.2 When slot welding is applied, the length of individual slots is to be not less than 75 mm with a minimum breadth of 2 times the plate thickness. The distance between ends of adjacent slots is to be not greater than 125 mm. The slots are to be fillet welded around the edges and filled with a suitable compound, e.g. epoxy putty. Slots are not to be filled with weld. The ends of slots are to be rounded.

2.4.3 The rudder, in way of rudder horn recesses of semi-spade rudders, are to have well radiused corners. The corner radii are not to be less than 5 times the local rudder plate thickness, but in no case less than 100 mm. Welding in the rudder side plating is to be positioned away from these corner radii. The weld connecting the side plate and the leading edge plate in way of these radiused corners is to be ground smooth.

2.4.4 Welds between plates and forged or cast parts or very thick plating are to be made as full penetration welds. In way of highly stressed areas e.g. cut-outs of semi-spade rudders and upper parts of spade rudders, cast or welded on ribs are to be arranged. Two sided full penetration welding is normally to be arranged. Where back welding is impossible welding is to be performed against ceramic backing bars or equivalent. Steel backing bars may be used and are to be continuously welded on one side to the heavy piece.

2.5 Equivalence

2.5.1 Lloyd's Register *(hereinafter referred to as LR)* may accept alternatives to the requirements given in this Section, provided they are deemed to be equivalent.

2.5.2 Direct analyses adopted to justify an alternative design are to take into consideration all relevant modes of failure, on a case by case basis. These failure modes may include, amongst others: yielding, fatigue, buckling and fracture. Possible damages caused by cavitation are also to be considered.

2.5.3 If deemed necessary by LR, lab tests, or full scale tests may be requested to validate the alternative design approach.

2.6 Rudder force

2.6.1 The lateral rudder force at the centre of pressure is to be determined for both ahead and astern conditions as follows:

$$C_R = 132\, K_1 K_2 K_3 A V^2 \text{ N}$$

where

A = rudder blade area, in m^2.

V = maximum service speed, in knots, for both the ahead and astern conditions.

= V_{ahead} is to be taken as the maximum design speed for short term high power operations. Where this speed is less than 10 knots, V_{ahead} is to be replaced by the following expression:

$$V_{min} = \frac{V_{ahead} + 20}{3}$$

= V_{astern}, is to be taken as the maximum astern speed or 0,5V_{ahead}, whichever is the greater.

K_1 = aspect ratio correction factor

= $\frac{\lambda + 2}{3}$ but is not to be taken greater than 2.

λ = $\frac{h_R^{\,2}}{A_t}$

h_R = mean height, in m, of the rudder blade, *see Figure 3.2.1 Rudder co-ordinate system*;

A_t = sum of rudder blade area A and area of rudder post or rudder horn, if any, within the mean height h_R, in m^2.

K_2 = rudder profile coefficient, *see Table 3.2.2 Rudder profiles*;

K_3 = 0,8 for rudders outside the propeller jet.

= 1,15 for rudders behind a fixed propeller nozzle.

= 1,0 otherwise.

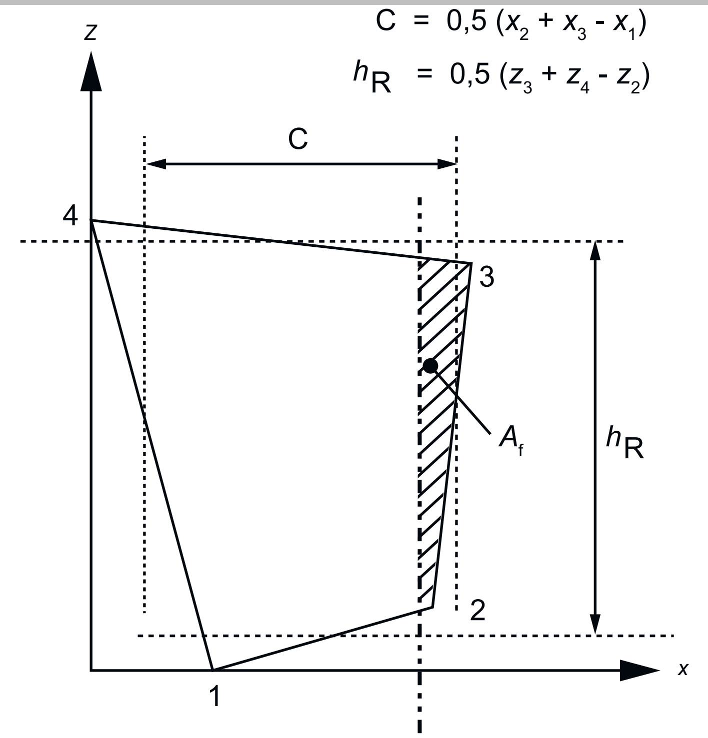
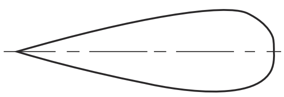
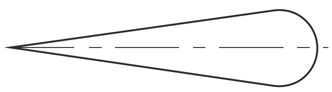
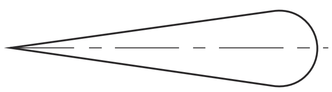
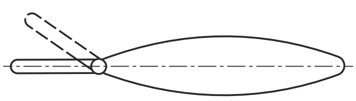
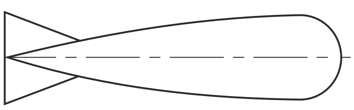
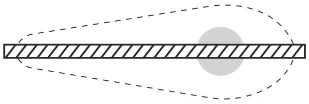


Figure 3.2.1 Rudder co-ordinate system

Table 3.2.2 Rudder profiles

Profile Type	*K2*	
	Ahead condition	Astern condition
NACA-00 series 	1,10	0,80
Flat sided 	1,10	0,90
Hollow 	1,35	0,90
High lift rudders 	1,70	To be specially considered
Fish tail 	1,40	0,80

Single plate 	1,00	1,00
Mixed profiles	1,21	0,90
Note For rudder profiles not defined above, the value of K_2 may be determined on the basis of experimental results. These results are to be submitted for consideration.		

2.7 Rudder torque for rudder blades without cut-outs

2.7.1 The maximum rudder torque, Q_R, is to be determined from both the ahead and astern conditions as follows:

Q_R = $C_R r$ Nm

where

C_R = lateral force acting on the rudder, as defined in *Vol 1, Pt 3, Ch 3, 2.6 Rudder force 2.6.1*.

r = distance from the centre of pressure to the centreline of the rudder stock.

= c (α – k_1), in m.

c = mean breadth of the rudder blade, (the mean chord length), in m, *see Figure 3.2.1 Rudder co-ordinate system*.

α = relative centre of pressure along the chord length, *see Table 3.2.3 Relative centre of pressure along the chord length, α*.

K_1 = ratio of the rudder blade area forward of the rudder stock centreline, to the rudder blade area:

= $\frac{A_f}{A}$

A_f = portion of the rudder blade area situated ahead of the centreline of the rudder stock.

For the ahead condition the rudder torque, Q_R is not to be taken less than:

Q_R = 0,1cC_R Nm

Table 3.2.3 Relative centre of pressure along the chord length, α

Condition	Behind fixed structure	Not behind a fixed structure
Ahead	0,25	0,33
Astern	0,55	0,66
Note Fixed structure is defined as any relatively stationary structure immediately ahead of the rudder, for example rudder horns of semi-spade rudders.		

2.8 Rudder torque for rudder blades with cut-outs (semi-spade rudders)

2.8.1 The maximum rudder torque, Q_R, is to be determined from both the ahead and astern conditions, for rudder blades with cut-outs, as follows. The pressure distribution for rudder blades with cut-outs is assumed to be proportional to the areas above and below the base of the cut-out. The rudder blade area, A, is to be divided into parts as per *Figure 3.2.2 Rudder Areas*.

Q_R = $Q_{R1} + Q_{R2}$ Nm

where

Q_{R1} = $C_{R1} r_1$.

Q_{R2} = $C_{R2} r_2$.

C_R = lateral force acting on the rudder, as defined in *Vol 1, Pt 3, Ch 3, 2.6 Rudder force 2.6.1*

C_{R1} = $C_R \frac{A_1}{A}$

C_{R2} = $C_R \frac{A_2}{A}$

r_1 = $c_1(\alpha - k_1)$, in m.

r_2 = $c_2(\alpha - k_2)$, in m.

c_1 = mean breadth, in m, of partial area A_1.

c_2 = mean breadth, in m, of partial area A_2.

α = relative centre of pressure along the chord length, *see Table 3.2.3 Relative centre of pressure along the chord length, α*.

K_1 = $\frac{A_{1f}}{A_1}$

K_2 = $\frac{A_{2f}}{A_2}$

For the ahead condition the rudder torque, Q_R is not to be taken less than:

$$Q_R = 0,1\frac{A_1c_1 + A_2c_2}{A}C_R \text{ Nm}$$

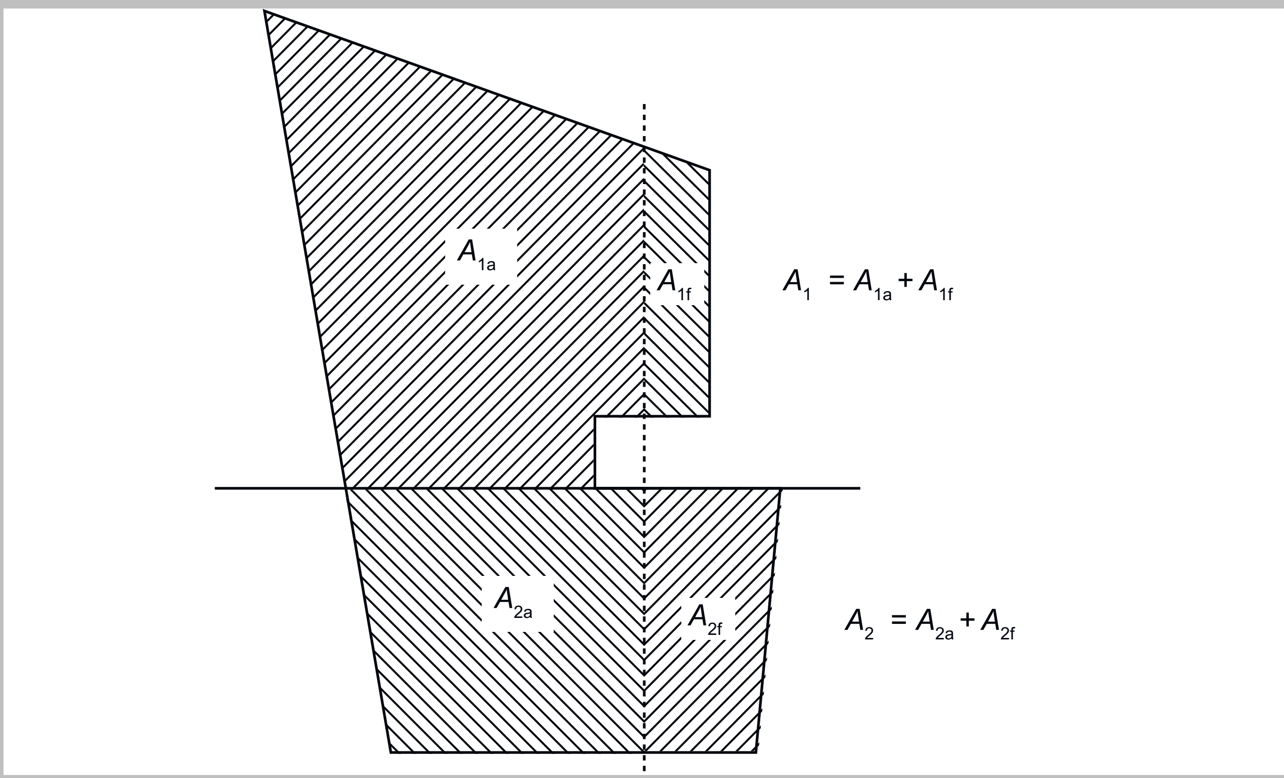


Figure 3.2.2 Rudder Areas

2.9 Rudder strength calculation

2.9.1 The rudder force and resulting rudder torques as given in *Vol 1, Pt 3, Ch 3, 2.6 Rudder force*, *Vol 1, Pt 3, Ch 3, 2.7 Rudder torque for rudder blades without cut-outs* or *Vol 1, Pt 3, Ch 3, 2.8 Rudder torque for rudder blades with cut-outs (semi-spade rudders)*, cause bending moments and shear forces in the rudder body, bending moments and torques in the rudder stock,

supporting forces in pintle bearings and rudder stock bearings and bending moments, shear forces and torques in rudder horns and heel pieces. The rudder body is to be stiffened by horizontal and vertical webs enabling it to act as a bending girder.

2.9.2 The bending moments, shear forces and torques as well as the reaction forces described in *Vol 1, Pt 3, Ch 3, 2.9 Rudder strength calculation 2.9.1* are to be determined by direct calculations or where otherwise stated by an approximate simplified formulae. For rudders supported by sole pieces or rudder horns these structures are to be included in the calculation model in order to account for the elastic support of the rudder body.

2.10 Rudder stock scantlings

2.10.1 The scantlings of the rudder stock are to be not less than required by *Table 3.2.4 Rudder stock diameter*.

2.10.2 The rudder stock diameter is to be dimensioned such that the stresses do not exceed the permissible stresses given in *Table 3.2.5 Rudder stock permissible stresses*.

2.10.3 Before significant reductions in rudder stock diameter due to the application of steels with yield stresses exceeding 235 N/mm^2 are granted, LR may require the evaluation of the rudder stock deformations. Large deformations of the rudder stock are to be avoided in order to avoid excessive edge pressures in way of bearings.

2.10.4 For spade rudders the stock diameter corrected for higher tensile steel is to be greater than 90 per cent of the uncorrected stock diameter unless direct calculations are submitted showing that the slope of the stock at the lowest main bearing does not exceed 0,0035 when the rudder blade is loaded by a lateral force of C_R, as defined in *Vol 1, Pt 3, Ch 3, 2.6 Rudder force 2.6.1,* acting at the centre of pressure.

2.10.5 For rudders having an increased diameter of the rudder stock in way of the rudder, the increased diameter is to be maintained to a point as far as practicable above the top of the lowest bearing. The diameter may then be tapered to the diameter required in way of the tiller. The length of the taper is to be at least three times the reduction in diameter. Particular care is to be taken to avoid the formation of a notch at the upper end of the taper, *see Figure 3.2.3 Rudder stock taper*.

2.10.6 Sudden changes of section or sharp corners in way of the rudder coupling, jumping collars and shoulders for rudder carriers, are to be avoided. Jumping collars are not to be welded to the rudder stock. Keyways in the rudder stock are to have rounded ends and the corners at the base of the keyway are to be radiused.

Table 3.2.4 Rudder stock diameter

Item	Requirement
(1) Rudder stock diameter due to combined loads	$d_c = d_t \sqrt[6]{1 + \frac{4}{3}\left(\frac{M}{Q_R}\right)^2}$
(2) Rudder stock diameter required for transmission of the rudder torque (e.g. in way of tiller)	$d_t = 4{,}2\sqrt[3]{Q_R k}$
Symbols	
Q_R = maximum rudder torque, in Nm, as calculated in *Vol 1, Pt 3, Ch 3, 2.8 Rudder torque for rudder blades with cut-outs (semi-spade rudders)* M = bending moment, in Nm, at the section of the rudder stock under consideration, *see Vol 1, Pt 3, Ch 3, 2.9 Rudder strength calculation 2.9.2*.	
Note 1. If direct calculations are not carried out, then the following approximate formulae may be applied: For rudders with a heel support; $M = \frac{h_R}{10\left(\frac{c^2}{A}\right)} C_R$ For spade rudders; $M = C_R h_c$ For semi-spade rudders;	

$$M = \frac{h_{\mathrm{R}}}{10\left(1 + \frac{c^2}{A}\right)} C_{\mathrm{R}}$$

where

h_{R} = mean height, in m, of the rudder blade, *see Figure 3.2.1 Rudder co-ordinate system*.

h_{c} = the distance, in m, from the centroid of the rudder blade area to the centre of the lowest main bearing;

c = mean breadth of the rudder blade, (the mean chord length), in m, *see Figure 3.2.1 Rudder co-ordinate system*.

A = rudder blade area, in m^2.

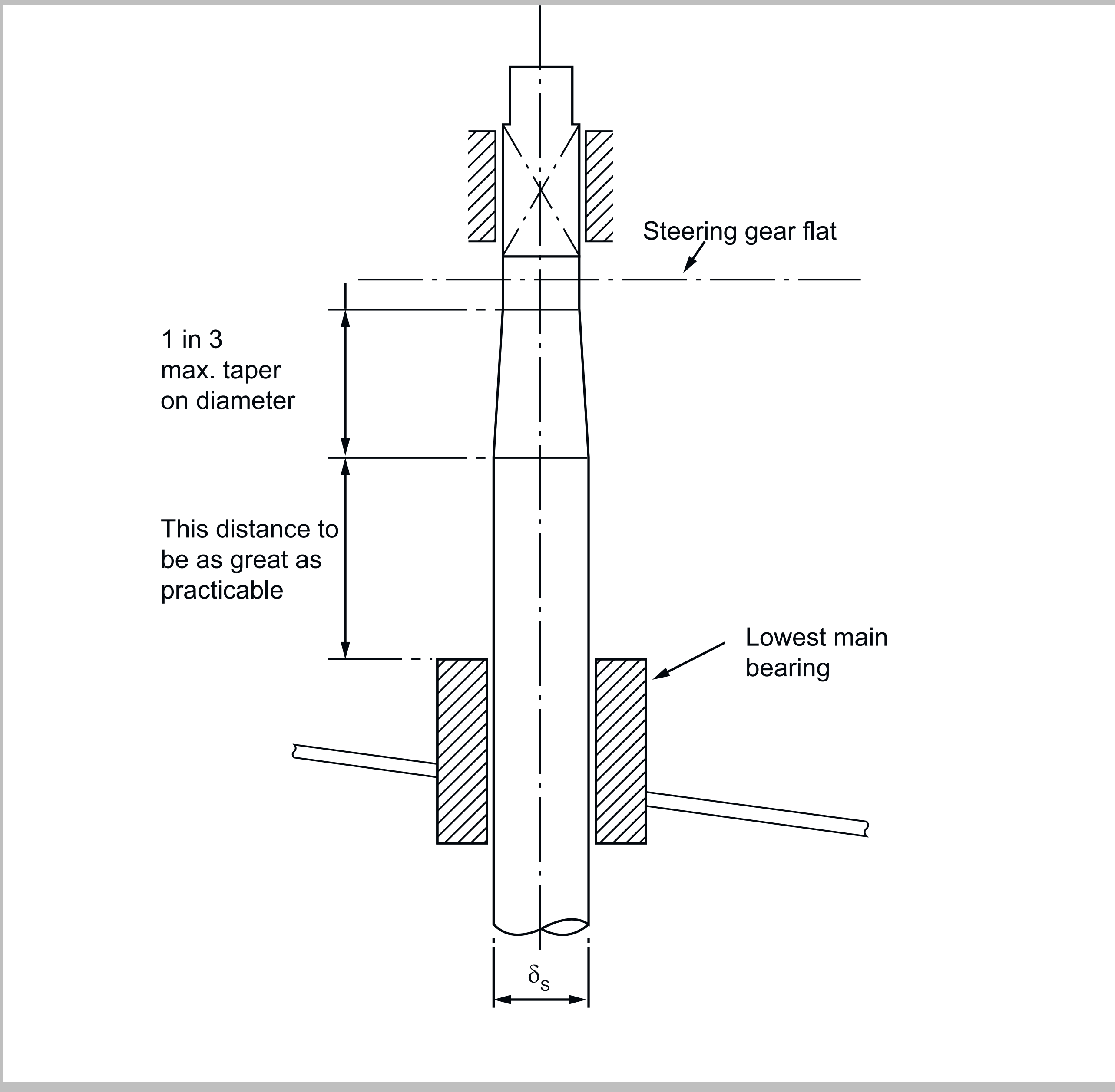


Figure 3.2.3 Rudder stock taper

Table 3.2.5 Rudder stock permissible stresses

Mode	Permissible stress, N/mm^2
(1) Torsional shear stress, τ_t	$\frac{68}{k}$
(2) Equivalent stress, σ_c	$\frac{118}{k}$
Symbols	
Equivalent stress: $\sigma_c = \sqrt{\sigma_b^2 + 3\tau_t^2}$ N/mm^2 Bending stress: $\sigma_b = 10,2 \times 10^3 \frac{M}{d_c^3}$ N/mm^2 Torsional stress: $\tau_t = 5,2 \times 10^3 \frac{Q_R}{d_c^3}$ N/mm^2 M = bending moment, in Nm, at the section of the rudder stock under consideration, *see Vol 1, Pt 3, Ch 3, 2.9 Rudder strength calculation 2.9.2*. d_c= actual stock diameter, in mm.	

2.11 Rudder blade

2.11.1 The scantlings of a double plated rudder are to be not less than required by *Table 3.2.6 Double plated rudder construction*.

2.11.2 The scantlings of single plate rudders are to be not less than required by *Table 3.2.7 Single plate rudder construction*.

2.11.3 All rudders are to be dimensioned such that the stresses do not exceed the permissible stresses given in *Table 3.2.8 Rudder blade permissible stresses*.

2.11.4 In way of rudder couplings, pintles, and cut-outs of semi-spade rudders the plating thickness is to be suitably increased. Adequate hand or access holes are to be arranged in the rudder plating in way of pintles as required, and the rudder plating is to be reinforced locally in way of these openings, *see Table 3.2.9 Thickness of side plating and vertical web plates in way of solid parts*.

2.11.5 For spade rudders fitted with a fabricated rectangular mainpiece, the mainpiece is to be designed with its forward and aft transverse sections at similar distances forward and aft of the rudder stock transverse axis.

2.11.6 Internal surfaces of double plate rudders are to be efficiently coated. Alternatively, where it is intended to fill the rudder with plastic foam or use a corrosion inhibitor, details are to be submitted. Means for draining the rudder are to be provided.

Table 3.2.6 Double plated rudder construction

Item		Requirement	
(1) Rudder side, top and bottom plating		$t = 5,5\ s\beta\sqrt{k\left(T + \frac{C_R 10^{-4}}{A}\right)} + 2,5$ mm	
(2) Webs, vertical and horizontal		$t_w \geq 0,7t$ but is not to be less than 8 mm	
(3) Nose plate		$t_n \geq 1,25t$ but need not exceed 22 mm	
(4) Mainpiece, (*see* Notes 1 and 2)	Rectangular (fabricated)	Breadth and width $\geq d_c$ The side plating of the mainpiece is to extend 0,2c and is to be in accordance with (1) and the vertical webs as per (2), but in no case are either to be less than t_M.	$t_M \geq 8,5 + 0,56\sqrt{d_c}\sqrt[3]{k}$ mm (*see* Notes 3 and 4)

	Tubular	Inside diameter ≥ d_c	

Symbols
T = draught, in m, as given in *Vol 1, Pt 3, Ch 1, 5.1 General*;
C_R = rudder force, in N, as defined in *Vol 1, Pt 3, Ch 3, 2.6 Rudder force 2.6.1*;
A = rudder area, in m^2;
$\beta = \sqrt{1,1 - 0,5\left(\frac{s}{b}\right)^2}$; max. 1,00 if b/s ≥ 2,5
k = material factor, as defined in *Vol 1, Pt 3, Ch 3, 2.3 Materials 2.3.4*
s = smallest unsupported width of plating in m
b = greatest unsupported width of plating in m
c = chord length in m, as defined in *Figure 3.2.1 Rudder co-ordinate system*.
Note 1. The mainpiece bending stresses are to be not greater than those in *Table 3.2.8 Rudder blade permissible stresses*. Note 2. The mainpiece plating attached to solid forged or cast parts, is not to be less than that required by *Vol 1, Pt 3, Ch 3, 2.12 Connections of rudder blade structure with solid parts*. Note 3. The stock diameter to be used for calculating the mainpiece plate thickness is to be based on the mild steel stock scantlings, as given in *Table 3.2.4 Rudder stock diameter*. Note 4. The requirement of t_M need only be applied to the upper part of the rudder plate: (a) for semi spade rudders; above a point midway between the lowest pintle and the bottom of the rudder. (b) for spade rudders; above a point one third of the height of the rudder above the base.

Table 3.2.7 Single plate rudder construction

Item	Requirement
(1) Blade thickness	The greater of; $t_b = 1,5\ sV\sqrt{k + 2,5}$; or 10 mm.
(2) Arms	$t_a = t_b$ The section modulus is not to be less than: $Z_a = 0,5\ s\ C_1^2\ V^2 k$ cm^3
(3) Mainpiece, *see* Note 1	As per *Table 3.2.4 Rudder stock diameter*
Symbols	
s = spacing of stiffening arms, in m, but is not to exceed 1 m. V = speed in knots, as defined in *Vol 1, Pt 3, Ch 3, 2.6 Rudder force 2.6.1*. C_1 = horizontal distance from the aft edge of the rudder to the centreline of the rudder stock, in m.	
Note 1. For spade rudders the lower third may be taper down to 0,75 d_c.	

Table 3.2.8 Rudder blade permissible stresses

Item	Permissible stress, N/mm^2		
	Bending stress, σ_b	Shear stress, τ	Equivalent stress, σ_c
Rudder blade, clear off cut-outs	$\frac{110}{k}$	$\frac{50}{k}$	$\frac{120}{k}$
Rudder blade in way of cut-outs, of semi-spade rudders	75	50	100

2.12 Connections of rudder blade structure with solid parts

2.12.1 Solid parts in forged or cast steel, which house the rudder stock or the pintle, are normally to be provided with protrusions, *see Figure 3.2.4 Cross-section of the connection between rudder blade structure and rudder stock housing*. These protrusions are not required when the web plate thickness is less than:

(a) 10 mm for web plates welded to the solid part on which the lower pintle of a semi-spade rudder is housed and for vertical web plates welded to the solid part of the rudder stock coupling of spade rudders.

(b) 20 mm for other web plates.

2.12.2 The solid parts are in general to be connected to the rudder structure by means of two horizontal web plates and two vertical web plates, *see Figure 3.2.4 Cross-section of the connection between rudder blade structure and rudder stock housing*.

2.12.3 The minimum section modulus of the cross-section of the structure of the rudder blade formed by vertical web plates and rudder plating, which is connected with the solid part where the rudder stock is housed is to be not less than:

$$w_s = c_s {d_c}^3 \left(\frac{H_{\mathrm{E}} - H_{\mathrm{x}}}{H_{\mathrm{E}}}\right) \frac{k}{k_{\mathrm{s}}} 10^{-4} \ \ \mathrm{cm}^3$$

where

c_{s} = coefficient, to be taken equal to:

= 1,0 if there is no opening in the rudder plating or if such openings are closed by a full penetration welded plate.

= 1,5 if there is an opening in the considered cross-section of the rudder.

d_{c} = rudder stock diameter, in mm.

H_{E} = vertical distance between the lower edge of the rudder blade and the upper edge of the solid part, in m.

H_{X} = vertical distance between the considered cross-section and the upper edge of the solid part, in m.

k = material factor for the rudder blade plating, *see Vol 1, Pt 3, Ch 3, 2.3 Materials 2.3.4*.

k_{s} = material factor for the rudder stock, *see Vol 1, Pt 3, Ch 3, 2.3 Materials 2.3.4*.

2.12.4 The actual section modulus of the cross-section of the structure of the rudder blade is to be calculated with respect to the symmetrical axis of the rudder, see the x-x axis in *Figure 3.2.4 Cross-section of the connection between rudder blade structure and rudder stock housing*. The breadth of the rudder plating to be considered for the calculation of section modulus is to be not greater than:

$$b = s_{\mathrm{v}} + \frac{2H_{\mathrm{x}}}{3} \qquad \mathrm{m}$$

where

s_{v} = spacing between the two vertical webs, in m.

Where openings for access to the rudder stock nut are not closed by a full penetration welded plate, they are not to be included in the section modulus calculations.

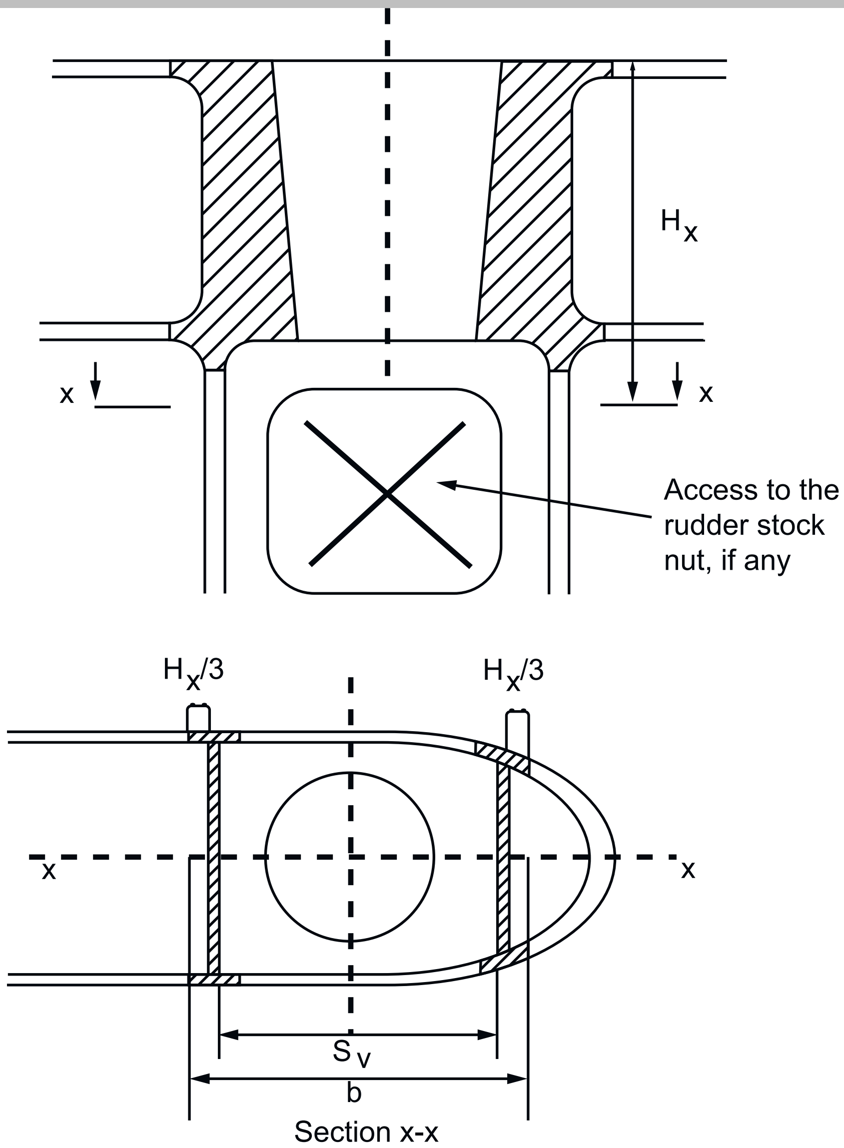


Figure 3.2.4 Cross-section of the connection between rudder blade structure and rudder stock housing

2.12.5 The thickness of the horizontal web plates connected to the solid parts as well as that of the rudder blade plating between these webs, is to be not less than the greater of the following values:

t_H = 1,2 t mm

t_H = $0,045 \dfrac{d_s{}^2}{S_H}$ mm

where

t = as calculated in *Table 3.2.6 Double plated rudder construction*

d_S = stock diameter, in mm, to be taken equal to:

= d_c, for the solid part housing the rudder stock, as calculated in *Table 3.2.4 Rudder stock diameter*.

= d_p, for the solid part housing the pintle, as calculated in *Table 3.2.12 Pintle requirements*.

S_H = spacing between the two horizontal web plates, in mm

The increased thickness of the horizontal webs is to extend fore and aft of the solid part at least to the next vertical web.

2.12.6 The thickness of the vertical web plates welded to the solid part where the rudder stock is housed as well as the thickness of the rudder side plating above and below this solid part is to be not less than the values obtained from *Table 3.2.9 Thickness of side plating and vertical web plates in way of solid parts*. The increased thickness is to extend above and below the solid piece at least to the next horizontal web.

Table 3.2.9 Thickness of side plating and vertical web plates in way of solid parts

Type of rudder	Thickness of vertical web plates, in mm		Thickness of rudder plating, in mm, *see* Note 1	
	Rudder blade without opening	Rudder blade with opening	Rudder blade without opening	Area with opening
Rudder supported by sole piece	1,2 t	1,6 t	1,2 t	1,4 t
Semi-spade and spade rudders	1,4 t	2,0 t	1,3 t	1,6 t
Symbols				
t = thickness of the rudder plating, in mm, as calculated in *Table 3.2.6 Double plated rudder construction* *c* = chord length in m, as defined in *Figure 3.2.1 Rudder co-ordinate system*.				
Note 1. The side plating of the mainpiece is to extend at least 0,2c.				

2.13 Rudder stock flange couplings

2.13.1 Rudder stock horizontal and vertical flange couplings are to be in accordance with *Table 3.2.10 Horizontal and Vertical flange couplings*.

2.13.2 For rudders with horizontal coupling arrangements the rudder stock should be forged when the stock diameter exceeds 350 mm. Where the stock diameter does not exceed 350 mm the rudder stock may be either forged or fabricated. Where the upper flange is welded to the rudder stock, a full penetration weld is required and its integrity is to be confirmed by non-destructive examination. The flange material is to be from the same welding materials group as the stock. Such rudder stocks are to be subjected to a furnace post-weld heat treatment (PWHT) after completion of all welding operations. For carbon or carbon manganese steels, the PWHT temperature is not to be less than 600°C.

2.13.3 For a horizontal flange coupling of a spade rudder, the palm radius, between the rudder stock and the flange, is not to be less than that calculated from *Figure 3.2.5 Rudder stock horizontal flange palm radius for spade rudders*, *see also Figure 3.2.7 Rudder stock horizontal flange coupling*.

where

d_c, R, t_f are defined in *Table 3.2.10 Horizontal and Vertical flange couplings*

b_f = breadth of flange, in mm.

2.13.4 For all rudder types, with a horizontal welded flange connection to the rudder stock, the connection details are in general to be in accordance with *Figure 3.2.6 Welded joint between rudder stock and coupling flange*.

2.13.5 The connecting bolts for coupling the rudder to the rudder stock are to be positioned with sufficient clearance to allow the fitting and removal of the bolts and nuts without contacting the palm radius. The surface forming the palm radius is to be free of hard and sharp corners and is to be machined smooth to the Surveyor's satisfaction. The surface in way of bolts and nuts is to be machined smooth to the Surveyor's satisfaction.

2.13.6 Coupling bolts are to be fitted bolts and their nuts are to be locked effectively.

Table 3.2.10 Horizontal and Vertical flange couplings

Item	Requirement	
	Horizontal coupling	Vertical coupling
Number of coupling bolts	$n \geq 6$	$n \geq 8$
Diameter of coupling bolts, in mm	$d_b = 0,62\sqrt{\frac{{d_c}^3 k_b}{n e_m k_s}}$	$d_b = 0,81 d_c \sqrt{\frac{k_b}{n k_s}}$
Thickness of coupling flange, in mm	The greater of the following, (*see* Note 1): (a) $t_f = d_b\sqrt{\frac{k_f}{k_b}}$ (b) $t_f = 0,9 d_b$ (c) $t_f = 0,33 d_c \sqrt[3]{k_s}$ (*see* Notes 2 and 3)	$t_f = d_b$
Width of flange material outside the bolt holes, in mm	$W_f = 0,67 d_b$	$W_f = 0,67 d_b$
First moment of area of bolts about centre of coupling, in cm^3	$m = 0,00071 n d_c {d_b}^2 \sqrt{\frac{k_b}{k_s}}$	$m = 0,00043 {d_c}^3 \sqrt{\frac{k_b}{k_s}}$
Stress concentration factor for as built scantlings	$\alpha_{asbuilt} \leq \alpha_{max}$ (*see* Note 2)	-
Symbols		

d_c = stock diameter, in mm, as calculated in *Table 3.2.4 Rudder stock diameter*.

n = total number of bolts.

e_m = mean distance, in mm, of the bolt axes from the centre of the bolt system.

k_s = material factor for the stock.

k_b = material factor for the bolts.

k_f = material factor for flange.

d_b = bolt diameter, in mm.

w_f = width of flange material outside the bolt holes, in mm.

m = first moment of area of bolts about the centre of the coupling, in cm^3.

$\alpha_{asbuilt}$ = stress concentration factor for as built scantlings.

$$\alpha_{asbuilt} = \frac{0,73}{\sqrt{\frac{R}{d}}}$$

α_{max} = maximum allowable stress concentration factor.

$$\alpha_{max} = \left(\left(53,82 - 35,29\, k_{max}\right)\frac{d^3}{h\, C_R\, 10^3}\right) - \left(\left(1,8 - 6,3\frac{R}{d}\right)\left(\frac{t_f - t_{fa}}{t_{fa}}\right)\right)$$

h = vertical distance, in m, between the centre of pressure and the centre point of the palm radius, *see Figure 3.2.7 Rudder stock horizontal flange coupling*.

k_{max} = the greater of k_s or k_f

R = palm radius, in mm, between the rudder stock and connection flange.

t_f = minimum thickness of coupling flange, in mm.

t_{fa} = as built flange thickness, in mm.

Note 1. Where the value of d_b is to be calculated for a number of bolts not exceeding 8.

Note 2. This requirement is only applicable for spade rudders with horizontal couplings, *see Figure 3.2.7 Rudder stock horizontal flange coupling*.

Note 3. For a twin spade rudder arrangement with a single screw, and where the rudders are positioned within the slipstream of the propeller:

(a) the thickness of the palm plate/coupling flange is not to be less than $0{,}35 d \sqrt[3]{k_s}$

(b) where the stock is welded to the palm plate, the stock diameter is to be increased by 14 per cent

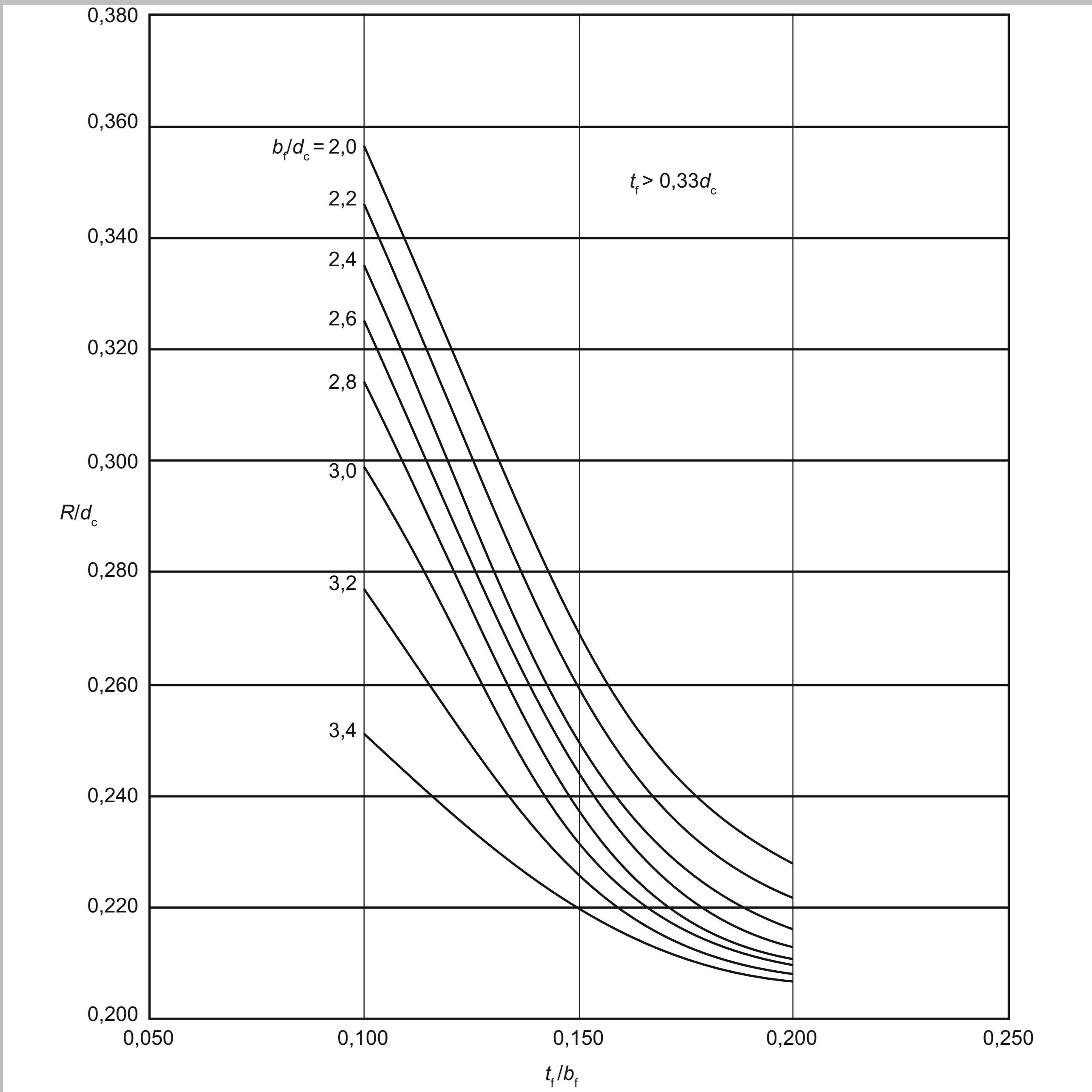


Figure 3.2.5 Rudder stock horizontal flange palm radius for spade rudders

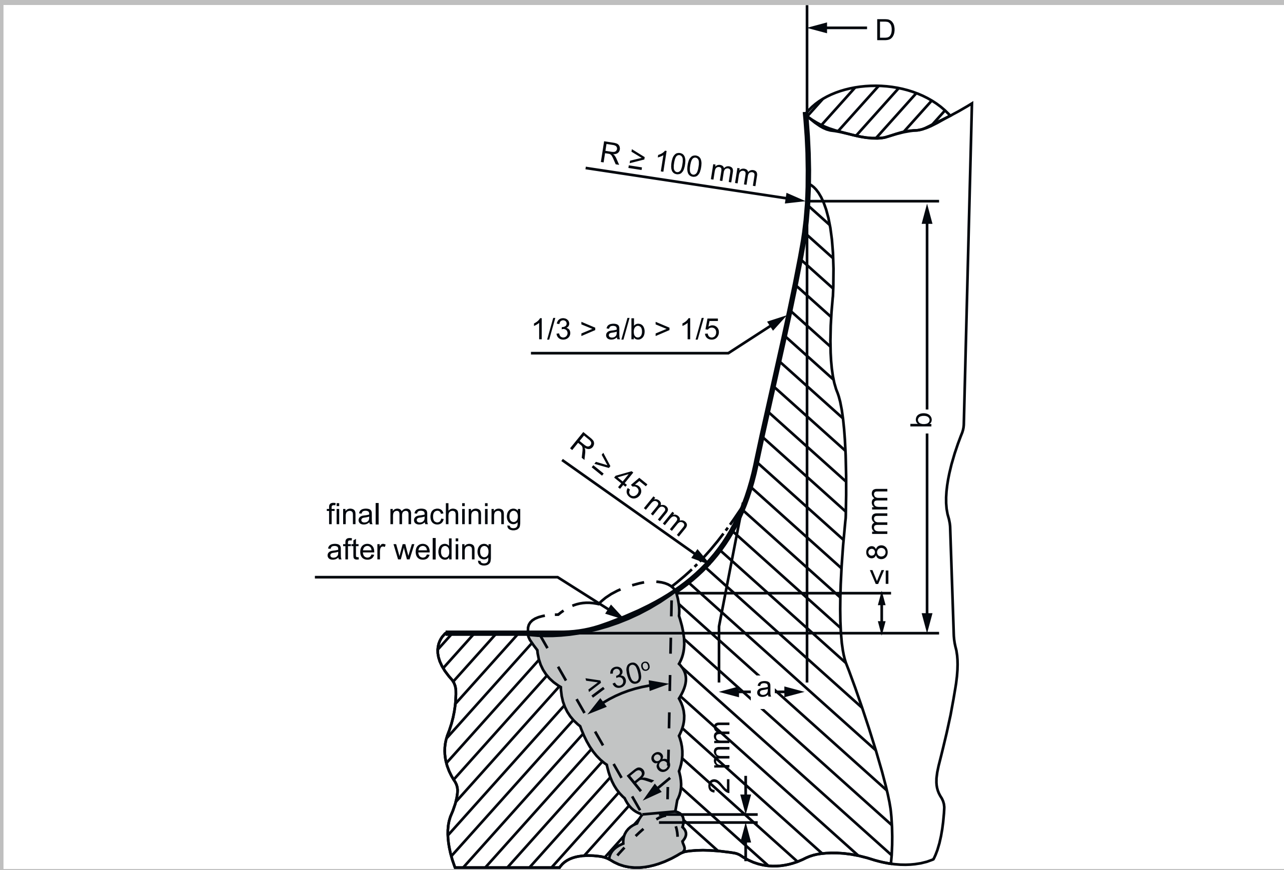


Figure 3.2.6 Welded joint between rudder stock and coupling flange

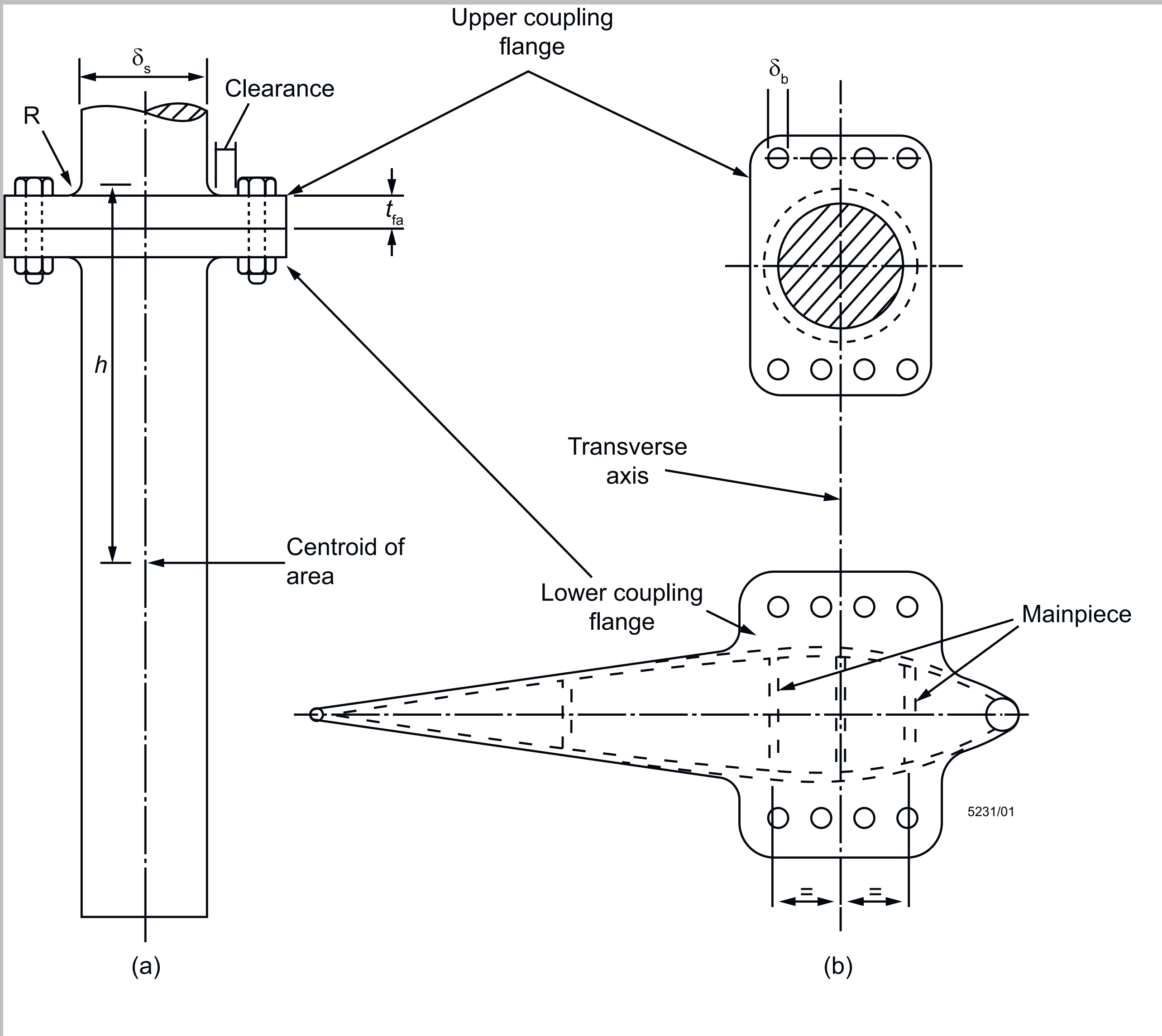


Figure 3.2.7 Rudder stock horizontal flange coupling

2.14 Cone couplings with key

2.14.1 Cone couplings without hydraulic arrangements for mounting and dismounting the coupling are to have a taper ratio, θ_t on diameter of 1:8 to 1:12;

where

$$\theta_t = \frac{d_c - d_u}{l}$$

The cone shapes are to fit exactly. The cone coupling is to be secured by a nut and the nut itself is to be secured, e.g. by a securing plate, *see Figure 3.2.8 Cone coupling with key*.

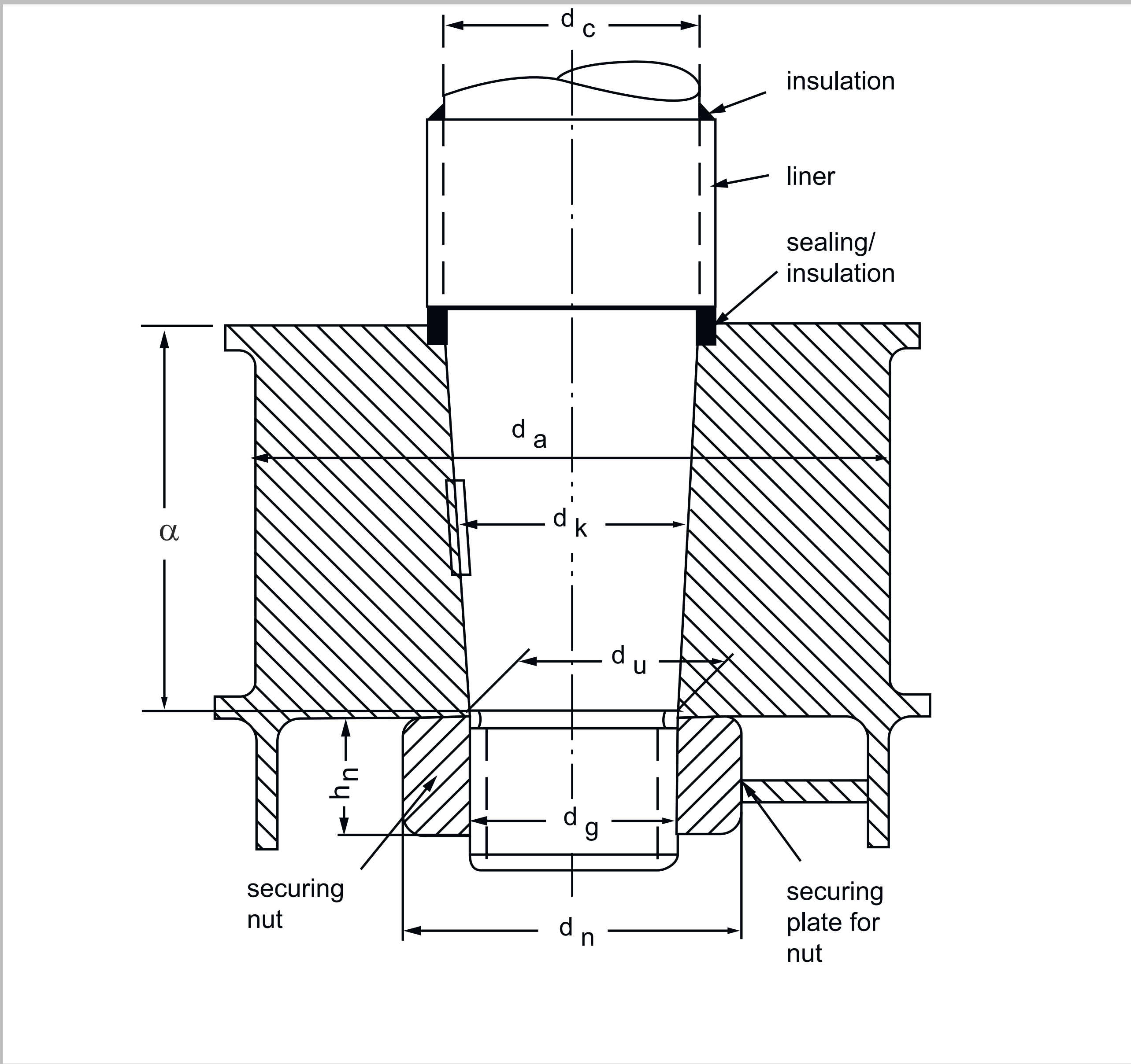


Figure 3.2.8 Cone coupling with key

2.14.2 The coupling length ℓ is to be, not less than $1{,}5 d_c$.

2.14.3 For couplings between stock and rudder where a key is provided, the shear area of the key is not to be less than:

$$a_s = \frac{17{,}55 Q_F}{d_k \, \sigma_{F1}} \quad \text{cm}^2$$

where

Q_F = design yield moment of rudder stock, in Nm.

$$Q_F = 0{,}02664 \frac{d_t^{\,3}}{k}$$

d_t = stock diameter, in mm, as calculated in *Table 3.2.4 Rudder stock diameter*.

d_k = mean diameter of the conical part of the rudder stock, in mm, at the key.

σ_{F1} = minimum specified yield stress of the key material, in N/mm^2.

Where the actual diameter d_{ta} is greater than the calculated diameter d_t, the diameter d_{ta} is to be used. However, d_{ta} need not be taken greater than 1,145 d_t.

2.14.4 The effective surface area of the key (without rounded edges) between key and rudder stock or cone coupling is not to be less than:

$$a_K = \frac{5Q_F}{d_k \ \sigma_{F2}} \quad \text{cm}^2$$

where

σ_{F2} = minimum specified yield stress of the key, stock or coupling material, in N/mm^2, whichever is less.

2.14.5 The dimensions of the securing nut are to be in accordance with *Table 3.2.11 Securing nut dimensions*, *see also Figure 3.2.8 Cone coupling with key*:

Table 3.2.11 Securing nut dimensions

Item	Requirement
External thread diameter	$d_g \geq 0{,}65\, d_c$
Height	$h_n \geq 0{,}6\, d_g$
Outer diameter	The greater of the following: (a) $d_n \geq 1{,}2\, d_i$, (b) $d_n \geq 1{,}5\, d_g$
Symbols, *see Figure 3.2.8 Cone coupling with key*.	
d_c = stock diameter, in mm d_g = external thread diameter, in mm h_n = height of securing nut, in mm d_n = minimum distance across flats of securing nut, in mm d_i = inner diameter of securing nut, in mm	

2.14.6 It is to be proved that 50 per cent of the design yield moment is solely transmitted by friction in the cone couplings. This can be done by calculating the required push-up pressure and push-up length in accordance with *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.4* and *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.5* for a torsional moment $Q'_F = 0{,}5Q_F$.

2.15 Cone couplings with special arrangements for mounting and dismounting the couplings

2.15.1 Where the stock diameter exceeds 200 mm, the press fit is recommended to be effected by a hydraulic pressure connection. In such cases the cone is to be more slender, and is to have a taper ratio, θ_t, on diameter of 1:12 to 1:20.

2.15.2 In the case of hydraulic pressure connections the nut is to be effectively secured against the rudder stock or the pintle.

2.15.3 For the safe transmission of the torsional moment by the coupling between rudder stock and rudder body the push-up pressure and the push-up length are to be determined in accordance with *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.4* and *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.5*.

2.15.4 The push-up pressure is not to be less than the greater of the two following values:

$$p_{req1} = \frac{2Q_F}{{d_m}^2 l \pi \mu_0} 10^3 \quad \text{N/mm}^2$$

$$p_{req2} = \frac{6M_b}{l^2\, d_m} 10^3 \quad \text{N/mm}^2$$

where

Q_F = design yield moment of rudder stock, in Nm, as defined in *Vol 1, Pt 3, Ch 3, 2.14 Cone couplings with key 2.14.3*.

d_m = mean cone diameter in, mm.

l = cone length in, mm.

μ_0 = frictional coefficient, to be taken as 0,15.

M_b = bending moment in the cone coupling (e.g. in case of spade rudders), in Nm.

It has to be proved by the designer that the push-up pressure does not exceed the permissible surface pressure in the cone. The permissible surface pressure is to be determined by the following formula:

$$p_{perm} = \frac{0,8\ \sigma_g\left(1-\alpha^2\right)}{\sqrt{3+\alpha^4}} \quad \mathrm{N/mm^2}$$

where

σ_g = minimum specified yield stress of the material of the gudgeon in N/mm^2.

α = $\dfrac{d_m}{d_a}$

d_m = mean cone diameter in, mm, *see Figure 3.2.8 Cone coupling with key*.

d_a = outer diameter of the gudgeon to be not less than 1,5 d_m, in mm, *see Figure 3.2.8 Cone coupling with key*.

2.15.5 The push-up length $\Delta\ell$, is to comply with the following formula, but in no case is to be less than 2 mm:

$\Delta\ell_1 \leq \Delta\ell \leq \Delta\ell_2$

where

$$\Delta\ell_1 = \frac{p_{req}\ d_m}{E\,\frac{1-\alpha^2}{2}\,\theta_t} + \frac{0,8\,R_{tm}}{\theta_t} \quad \mathrm{mm}$$

$$\Delta\ell_2 = \frac{1,6\,\sigma_g d_m}{E\theta_t\sqrt{3+\alpha^4}} + \frac{0,8\,R_{tm}}{\theta_t} \quad \mathrm{mm}$$

where

P_{req} σ_g, α, d_m are defined in *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.4*

R_{tm} = mean roughness, in mm, taken equal to 0,01

E = Young's modulus of the material, in N/mm^2

θ_t = taper on diameter, *see Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.1*

2.15.6 In case of hydraulic pressure connections the required push-up force P_e, for the cone may be determined by the following formula:

$$P_e = p_{req} d_m \Pi\ \ell \frac{\theta_t}{2} + 0,02 \quad \mathrm{N}$$

The value 0,02 is a reference for the friction coefficient using oil pressure. It varies and depends on the mechanical treatment and roughness of the details to be fixed. Where due to the fitting procedure a partial push-up effect caused by the rudder weight is given, this may be taken into account when fixing the required push-up length, subject to approval by LR.

2.16 Pintles

2.16.1 Rudder pintles and their bearings are to be in accordance with the requirements of this sub-section and *Vol 1, Pt 3, Ch 3, 2.17 Bearings*.

2.16.2 The bottom pintle on semi-spade rudders and all pintles over 500 mm in diameter are if inserted into their sockets from below, to be keyed to the rudder or sternframe as appropriate or to be hydraulically assembled, with the nut adequately locked, *see Vol 1, Pt 3, Ch 3, 2.14 Cone couplings with key* and *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings*.

2.16.3 Where liners are fitted to pintles, they are to be shrunk on or otherwise efficiently secured. If liners are to be shrunk on, the shrinkage allowance is to be indicated on the plans. Where liners are formed by stainless steel weld deposit, the pintles are to be of weldable quality steel and details of the procedure are to be submitted, *see also Vol 1, Pt 3, Ch 3, 2.17 Bearings 2.17.2*.

2.16.4 Where an ***IWS** (In-water Survey) notation is to be assigned, means are to be provided for ascertaining the rudder pintle and bush clearances and for verifying the security of the pintles in their sockets with the vessel afloat.

Table 3.2.12 Pintle requirements

Item	Requirement	
(1) Pintle diameter, in mm	$d_{\text{p}} = 0,35\sqrt{B\,k_{\text{p}}}$	
(2) Pintle taper	Method of assembly	Taper (on diameter)
	Keyed and other manually assembled pintles applying locking by securing nut	1:8 – 1:12
	Pintles mounted with oil injection and hydraulic nut	1:12 – 1:20
(3) Pintle bearing length	$d_{\text{p}} \leq l_{\text{p}} \leq 1,2\ d_{\text{p}}$	
(4) Pintle housing/gudgeon	$b_{\text{g}} \geq 0,25\ d_{\text{p}}$	
(5) Liner or bush in way of pintle bearings, in mm	$t = 0,01\sqrt{B}$	
Symbols		
k_{p} = material factor for pintle B = bearing force, in d_{p} = actual pintle diameter measured on the outside of liners. b_{g} = thickness of pintle housing/gudgeon in way of pintles (measured from the outside of bush if fitted).		
Note 1. The length of the pintle housing in the gudgeon is not to be less than the maximum pintle diameter d_{p}. Note 2. The minimum dimensions of threads and nuts are to be determined according to *Table 3.2.11 Securing nut dimensions*.		

2.16.5 The required push-up pressure for pintle bearings is to be determined by the following formula:

$$p_{\text{req}} = 0,4\frac{B d_{\text{p}}}{d_{\text{m}}^2\ \ell} \quad \text{N/mm}^2$$

where

d_{m}, and l are defined in *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.4*

B = supporting force in the pintle bearing, in N

d_{p} = pintle diameter, in mm.

The push-up length is to be calculated in accordance with *Vol 1, Pt 3, Ch 3, 2.15 Cone couplings with special arrangements for mounting and dismounting the couplings 2.15.5*, using the required push-up pressure and properties for the pintle bearing.

2.17 Bearings

2.17.1 Bearings are to comply with the requirements of *Table 3.2.13 Bearings*. The fitting of bearings is to be carried out in accordance with the manufacturer's recommendations to ensure that they remain secure under all foreseeable operating conditions.

2.17.2 Where it is proposed to use stainless steel for liners or bearings for rudder stocks and/or pintles, the chemical composition is to be submitted for approval. Synthetic rudder bearing materials are to be of a type approved by LR. When this type of lining material is used, arrangements to ensure an adequate supply of sea-water to the bearing are to be provided.

Table 3.2.13 Bearings

Item	Requirement
(1) Bearing surface area	$A_{\text{B}} = \frac{B}{q_{\text{a}}}$ mm^2
(2) Bearing length	The length/diameter ratio of the bearing surface is not to be greater than 1,2.

(3) Clearance	Bearing material	Minimum clearance (on diameter)
	Metal	0,001d + 1,0
	Synthetic	*See* Notes 1, 2, 3, and 4
(4) Liners and bushes	Material	Minimum thickness
	Metal and synthetic material	8 mm
	Lignum vitae	22 mm
(5) Main bearing wall thickness, *see* Note 5	Greater than 0,2d_c	
Symbols		
A_B = bearing surface, in mm^2, defined as the projected area (length x outer diameter) of liner d = stock diameter, as calculated in *Table 3.2.4 Rudder stock diameter*, or pintle diameter as calculated in *Table 3.2.12 Pintle requirements* B = bearing force, in N. q_a = allowable surface pressure, *see Table 3.2.14 Maximum surface pressure*.		
Note 1. If non-metallic bearing material is applied, the bearing clearance is to be specially determined considering the material's swelling and thermal expansion properties. This clearance is not to be less than 1,5 mm on bearing diameter unless a smaller clearance is supported by the manufacturer's recommendation and there is documented evidence of satisfactory service history with a reduced clearance. Note 2. For bearings which are pressure lubricated the clearance must be restricted to enable the pressure to be maintained. Note 3. The value of the proposed minimum clearance is to be indicated on plans submitted for approval. Note 4. Proposals for higher pressures or other materials will be specially considered on the basis of satisfactory test results. Note 5. Where web stiffening is fitted on the bearing, a reduction in wall thickness will be considered.		

Table 3.2.14 Maximum surface pressure

Bearing material	q_a (N/mm^2) (*see* Note 1)
Lignum vitae	2,5
White metal, oil lubricated	4,5
Synthetic material with hardness between 60 and 70 Shore D (*see* Note 2)	5,5 (*see* Note 3)
Steel (*see* Note 4) and bronze and hot-pressed bronze-graphite materials	7,0
Note 1. Proposals for higher pressures will be specially considered on the basis of satisfactory test results. Note 2. Indentation hardness test at 23°C and with 50 per cent moisture according to a recognised standard. Synthetic bearing materials are to be of an approved type. Note 3. Surface pressures exceeding 5,5 N/mm^2 may be accepted in accordance with bearing manufacturer's specification and tests, but in no case more than 10 N/mm^2. Note 4. Stainless and wear-resistant steel in an approved combination with stock liner.	

■ *Section 3* Stabiliser arrangements

3.7 Fin stock diameter in way of tiller, d_{Fu}

3.7.1 The fin stock diameter in way of the tiller, d_{Fu}, is to be not less than that determined from the formula:

$$d_{FU} = 42\sqrt[3]{Q_F k}\ \text{mm}$$

$$\cancel{d_{FU} = 42\sqrt[3]{\frac{Q_F}{K_o}}\ \text{mm}}$$

where

Q_F = fin torque (in the appropriate condition), in kNm, as given in 3,5

~~K~~ k_o = material factor, as defined in ~~*Vol 1, Pt 3, Ch 3, 1.4 Materials 1.4.3*~~ *Vol 1, Pt 3, Ch 3, 2.3 Materials 2.3.4*

3.8 Fin stock diameter, d_F

3.8.1 For a fin stock subjected to combined torque and bending, the equivalent stress in the fin stock is not to exceed that determined from the following:

= ~~$\sigma_e \leq 118 K_o$ N/mm^2~~

$$\sigma_e \leq \frac{118}{k}\ \text{N/mm}^2$$

where

K_o = ~~material factor, as defined in *Vol 1, Pt 3, Ch 3, 1.4 Materials 1.4.3*.~~

k = material factor, as defined in *Vol 1, Pt 3, Ch 3, 2.3 Materials 2.3.4*

The equivalent stress is to be determined by the formula:

$$\sigma_e = \sqrt{\sigma_b^2 + 3\tau_t^2}\ \text{N/mm}^2$$

Bending stress: = $\sigma_b = 10200 \dfrac{M_F}{d_F^3} \times 10^3\ \text{N/mm}^2$

Torsional stress: = $\tau_t = 5100 \dfrac{Q_R}{d_F^3} \times 10^3\ \text{N/mm}^2$

3.8.2 The basic fin stock diameter, d_F, at and below the lowest bearing is not to be less than that determined from the following:

$$\cancel{d_F = d_{Fu}\sqrt[6]{1 + \frac{4}{3}\left(\frac{M_F}{Q_F}\right)^2}\ \text{mm}}$$

$$d_F = d_{Fu}\sqrt[6]{1 + \frac{4}{3}\left(\frac{M_F}{Q_F}\right)^2}\ \text{mm}$$

where

d_{Fu} = diameter of the fin stock in way of the tiller, in mm

M_F = fin bending moment, kNm, *see Vol 1, Pt 3, Ch 3, 3.6 Fin bending moment, MF*

Q_F = fin torque (in the appropriate condition), in kNm, as given in *Vol 1, Pt 3, Ch 3, 3.5 Fin torque, QF*

3.9 Fin plating

3.9.1 The thickness of the fin side plating is not to be less than that determined from the following:

$$\cancel{t = 0{,}0224\, s\, \beta \sqrt{\frac{P_F}{110 K_o}} + 2{,}5 \text{ mm}}$$

$$t = 0{,}0224 s \beta \sqrt{\frac{P_F k}{110}} + 2{,}5 \text{ mm}$$

where

s = stiffener spacing, in mm

β = panel aspect ratio correction factor

= $A_R (1 - 0{,}25 A_R)$ for $A_R \leq 2$

= 1 for $A_R > 2$

A_R = panel aspect ratio

= panel length/panel breadth

P_F = fin pressure, in kN/m^2

= $10T + \frac{F_F}{A_F}$ kN/mm^2

T = maximum draught, in metres

F_F = fin force, in kN, *see Vol 1, Pt 3, Ch 3, 3.4 Fin force, FF*

A_F = fin area, in m^2

~~K_o = material factor, as defined in *Vol 1, Pt 3, Ch 3, 1.4 Materials 1.4.3*.~~

k = material factor, as defined in *Vol 1, Pt 3, Ch 3, 2.3 Materials 2.3.4*

3.9.2 The thickness of the nose plates is not to be less than 1,25 times the thickness of the fin side plating. The thickness of web plates is not to be less than 70 per cent of the thickness of the fin side plating, or 6 mm, whichever is the greater.

3.9.3 Alternative materials and methods for fin stabilisers will be specially considered.

■ *Section 5*
Fixed and steering nozzles, bow and stern thrust units, ducted propellers

5.4 Ancillary items

5.4.1 The diameter of pintles and the diameter and first moment of area about the stock axis of coupling bolts are to be derived from ~~*Vol 1, Pt 3, Ch 3, 2.23 Pintles*~~ *Vol 1, Pt 3, Ch 3, 2.16 Pintles* and ~~*Vol 1, Pt 3, Ch 3, 2.24 Bolted couplings*~~ *Vol 1, Pt 3, Ch 3, 2.13 Rudder stock flange couplings* respectively.

5.4.2 Suitable arrangements are to be provided to prevent the steering nozzle from lifting.

Volume 1, Part 3, Chapter 4
Closing Arrangements and Outfit

■ *Section 7*
Air pipes

7.2 Height of air pipes

7.2.1 The height of air pipes from the upper surface of decks exposed to the weather, to the point where water may have access below is normally to be not less than:

- 760 mm on exposed decks immediately above the design draught, e.g. quarter decks and well decks.
- 450 mm measured above deck sheathing, where fitted elsewhere.

7.2.2 Lower heights may be approved in cases where these are essential for the working of the ship, provided that the design and arrangements are otherwise satisfactory. In such cases, efficient, permanently attached closing appliances as required by *Vol 1, Pt 3, Ch 4, 7.3 Closing appliances 7.3.1* are to be of an approved automatic type.

7.2.3 The height of air pipes may be required to be increased on ships where this is shown to be necessary by the stability and watertight subdivision calculations required by *Vol 1, Pt 1, Ch 1, 1 Background 1.1* . An increase in height may also be required when air pipes to fuel oil and settling tanks are situated in positions where sea water could be temporarily entrapped, e.g. in recesses in the sides and ends of superstructures or deckhouses, between hatch ends, behind high sections of bulwark, etc.. This may entail an increase in tank scantlings, *see also Vol 1, Pt 6, Ch 3 Scantling Determination*.

7.2.4 Air pipes are generally to be led to an exposed deck. *See also Vol 2, Pt 7, Ch 2, 10.4 Termination of air pipes 10.4.4*.

7.2.5 Where air pipes are led through the side of superstructures, the opening is to be at least 2,3 m above the design waterline.

7.2.6 The minimum wall thickness of air pipes in positions indicated in *Vol 1, Pt 3, Ch 4, 7.2 Height of air pipes 7.2.1* is to be:

- 6,0 mm for pipes of 80 mm external diameter or smaller.
- 8,5 mm for pipes of 165 mm external diameter or greater.

(a)

Intermediate minimum thicknesses are to be determined by linear interpolation.

7.2.7 Air pipe coaming heights may be reduced on ships assigned a service area notation **SA4**. Coaming heights are to be as high as practicable, with a minimum height of 300 mm.

Volume 1, Part 4, Chapter 3
Special Features

■ *Section 1*
General

1.1 Application

1.1.1 The requirements of this Chapter are applicable to mono-hull and multi-hull ships of steel construction as defined in *Vol 1, Pt 1, Ch 1, 1 Background* .

Volume 1, Part 5, Chapter 4
Global Design Loads

■ *Section 6*
Loading guidance information

6.1 General

6.1.1 Sufficient information is to be supplied to every ship to enable the Commanding Officer or Master to arrange loading and ballasting in such a way as to avoid the creation of unacceptable stresses in the ship's structure.

6.1.2 This information should include any limiting criteria that were applied to the structural design of the ship, e.g. the maximum and minimum still water bending moments, service area notation with details of service area restrictions.

6.1.3 This information is to be provided by means of a Stability Information Book or Loading Manual and in addition, where required, by means of an approved loading instrument.

6.1.4 An Operational manual which contains the ship's assigned operational envelope is to be provided on board, *see Vol 1, Pt 1, Ch 2, 2 Scope of the Rules* and *Vol 1, Pt 1, Ch 1, 1 Background* .

Volume 1, Part 6, Chapter 3
Scantling Determination

■ *Section 4*
NS2 and NS3 scantling determination

4.7 Deck structures

4.7.1 The requirements of this Section are applicable to longitudinally and transversely framed deck structure.

4.7.2 The thickness of the deck plating is in no case to be less than the appropriate minimum requirement given in *Vol 1, Pt 6, Ch 3, 2 Minimum structural requirements*

4.7.3 Additional requirements for deck structures are indicated in *Vol 1, Pt 6, Ch 3, 10 Deck structures*

4.7.4 The thickness requirement for deck plating may be determined from the general equations given in *Vol 1, Pt 6, Ch 2, 2.7 Plating general*, the pressures given in *Table 3.4.5 Deck structures* and the structural design factors in *Vol 1, Pt 6, Ch 5 Structural Design Factors*

4.7.5 The section modulus, inertia and web area requirements for deck stiffening may be determined from the general equations given in *Vol 1, Pt 6, Ch 2, 2.8 Stiffening general*, the pressures given in *Table 3.4.5 Deck structures*and the structural design factors in *Vol 1, Pt 6, Ch 5 Structural Design Factors*

Table 3.4.5 Deck structures

Structural element	Design pressure	Beam model	Stiffening type factor, δ_f	Remarks
(1) Weather decks and exposed decks				
Plating	The greater of (a) $1,26P_{wd}$ (b) P_{cd}	—	—	—
Secondary stiffening Deck longitudinals or deck beams	The greater of (a) δ_f $1,26P_{wd}$ (b) P_{cd}	B	0,8	*See* Note 1
Primary stiffening Deck girders or deck transverses or deep beams	The greater of (a) δ_f $1,26P_{wd}$	A	0,5	*See* Note 2

	(b) P_{cd}			
(2) Lower decks and internal decks				
Plating	The greater of (a) P_{in} (b) P_{cd}	—	—	—
Secondary stiffening Deck longitudinals or deck beams	The greater of (a) P_{in} (b) P_{cd}	B	—	*See* Note 1
Primary stiffening Deck girders or deck transverses or deep beams	The greater of (a) P_{in} (b) P_{cd}	A	—	*See* Note 2
(3) Ramps and lifts				
Plating	P_{ra}	—	—	—
Secondary stiffening Deck longitudinals or deck beams	P_{ra}	B	—	*See* Note 1
Primary stiffening Deck girders or deck transverses or deep beams	P_{ra}	E	—	*See* Note 3

Symbols

P_{wd} = the pressure acting on exposed and weather decks, as defined in *Vol 1, Pt 5, Ch 3, 3.5 Pressure on exposed and weather decks, Pwd*, *see also Vol 1, Pt 5, Ch 3, 5.2 Pressure on external decks*

P_{in} = the pressure acting on internal decks, as defined in *Vol 1, Pt 5, Ch 3, 5.3 Pressure on internal decks, Pin*

P_{cd} = the pressure acting on decks designed to carry cargo or heavy equipment loads, as defined in *Vol 1, Pt 5, Ch 3, 5.4 Loads for decks designed for cargo or heavy equipment loads, Pcd and Wcd*, where appropriate

P_{ra} = the pressure on ramps and lifts, as defined in *Vol 1, Pt 5, Ch 3, 6.2 Loads for ramps and lifts, Pra*

Note 1. In general, secondary stiffening members are to be designed using load model 'B', *see Table 2.2.1 Section modulus, inertia and web area coefficients for different load models*. Such members are in general to be continuous or made effectively continuous by means of suitable bracketing.

Note 2. Guidelines for the design of primary stiffening members are given in ~~*Table 3.2.1 Rudder profile coefficient,* K_2 In general, primary stiffening members are to be designed using load model 'A',~~ *Table 3.2.* In general, primary stiffening members are to be designed using load model 'A', *see Table 2.2.1 Section modulus, inertia and web area coefficients for different load models*. Primary members are to be substantially bracketed at their end connections.

Note 3. In general, primary lift and ram stiffening members are to be designed using load model 'E', *see Table 2.2.1 Section modulus, inertia and web area coefficients for different load models*

Note 4. Where a deck forms the boundary of a deep tank, the deck is to be examined for compliance with the requirements for deep tanks, *see Table 3.4.4 Watertight and deep tank bulkhead scantlings*

Note 5. Where a deck forms the boundary of a watertight subdivision or part of the shell envelope, the deck is to be examined for compliance with the requirements for watertight bulkheads, *see Table 3.4.4 Watertight and deep tank bulkhead scantlings*, or the shell envelope respectively, *see Table 3.4.2 Shell envelope framing*

Note 6. Where a deck is subject to wheel loadings arising from vehicles or helicopters/aircraft, such decks are to be examined for compliance with the requirements for vehicle decks, *see Vol 1, Pt 4, Ch 3, 2 Vehicle decks and fixed ramps*, or aircraft operation, *see Vol 1, Pt 4, Ch 2, 10 Aircraft operations*, as appropriate.

■ *Section 8*
Double bottom structures

8.1 General

8.1.1 The requirements of this Section, unless specified otherwise, are applicable to all ship types, NS1, NS2 and NS3.

8.1.2 The basic structural scantlings of NS1 ships are to be determined in accordance with *Vol 1, Pt 6, Ch 3, 3 NS1 scantling determination* The basic structural scantlings of NS2 and NS3 ships are to be determined in accordance with *Vol 1, Pt 6, Ch 3, 4 NS2 and NS3 scantling determination*. The scantlings of the double bottom structure are also to comply with the appropriate minimum requirements given in *Vol 1, Pt 6, Ch 3, 2 Minimum structural requirements* The arrangements of double bottoms are to comply with the requirements of *~~Vol 1,~~ Pt 3, Ch ~~3, 2.6 Rudder profile coefficient, K2~~ 3,2.6.*

8.1.3 Where a double bottom is required to be fitted, its depth at the centreline, d_{DB}, is to be in accordance with *Vol 1, Pt 6, Ch 3, 2 Minimum structural requirements*

8.1.4 This Section provides for longitudinal or transverse framing in the double bottom, but for NS1 and NS2 ships longitudinal framing is in general to be adopted. *See Vol 1, Pt 3, Ch 2, 3.1 General*

8.1.5 Where appropriate all ship types are to comply with the requirements of *Vol 1, Pt 6, Ch 3, 14 Strengthening for bottom slamming* for bottom slamming.

8.1.6 Where a floor, girder or inner bottom plating forms the boundary of a tank or part of the watertight subdivision, the requirements of *Vol 1, Pt 6, Ch 3, 9 Bulkheads and deep tanks* for deep tanks and watertight bulkheads are to be complied with. For corrosion margins, *see Vol 1, Pt 6, Ch 6, 2.10 Corrosion margin*

Volume 1, Part 6, Chapter 6
Material and Welding Requirements

■ *Section 1*
General

1.1 Application

1.1.1 The requirements of this Chapter are applicable to mono-hull ships of steel construction as defined in *Vol 1, Pt 1, Ch 1, 1 Background* .

Section 4
Design and construction

4.3 Azimuth thrusters with a nozzle

4.3.1 Where the propeller is contained within a nozzle, the equivalent rudder stock diameter in way of tiller, used in ~~*Table 3.2.12 Rudder couplings to stock* in Vol 1, Pt 3, Ch 3, is to be determined as follows:~~ *Table 3.2.4 Rudder stock diameter* in *Vol 1, Pt 3, Ch 3 Ship Control Systems*, is to be determined as follows:

$$\cancel{d_{su} = 26,03\sqrt[3]{(V+3)^2 A_N X_{PF}}\ \text{mm}}$$

$$d_{su} = 26,03\sqrt[3]{(V+3)^2 A_N X_{PF}}\ \text{mm}$$

where

V = maximum service speed, in knots, which the craft is designed to maintain under thruster operation

A_N = projected nozzle area, in m^2, and is equal to the length of the nozzle multiplied by the mean external vertical height of the nozzle

X_{PF} = horizontal distance from the centreline of the steering tube to the centre of pressure, in metres. The position of the centre of pressure is determined from *Table ~~3.2.7 Single~~ .Single plate rudder construction* in *Vol 1, Pt 3, Ch 3 Ship Control Systems*

The corresponding maximum turning moment, M_T, is to be determined as follows:

M_T = 11,1 x d_{SU}^3 Nmm.

4.3.2 In addition to the requirements of *Vol 1, Pt 3 Design Principles and Constructional Arrangements* the scantlings of the nozzle stock or steering tube are to be such that the section modulus Z against transverse bending at any section x–x is not less than:

$$Z = 1,73\sqrt{(V+3)^4 A_N{}^2 X_{PF}{}^2 + \frac{a^2}{4} T_M{}^2 10^4}\ \text{cm}^3$$

where

a = dimension, in metres, as shown in *Figure 3.4.1 Azimuth thruster*

T_M = maximum thrust of the thruster unit, in tonnes.

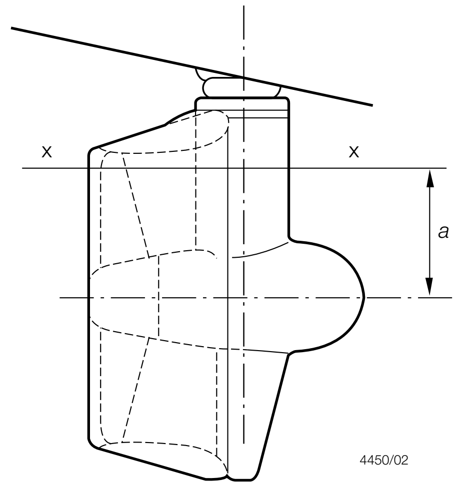


Figure 3.4.1 Azimuth thruster

4.3.3 The scantlings of nozzle connections or struts will be specially considered. In the case of certain high powered ships, direct calculation may be required.

4.3.4 Where the propeller is not contained in a nozzle, the scantlings in way of the tiller will be subject to special consideration.

Volume 2, Part 4, Chapter 4 Podded Propulsion Units

■ *Section 5* Structure design and construction requirements

5.1 Pod structure

5.1.1 Podded unit struts and pod bodies may be of cast, forged or fabricated construction or a combination of these construction methods.

5.1.2 Means are to be provided to enable the propeller shaft, bearings and seal arrangements to be examined in accordance with LR's requirements and the manufacturer's recommendations.

5.1.3 When high tensile steel fasteners are used as part of the structural arrangement and there is a risk that these fasteners may come into contact with sea-water, carbon-manganese and low alloy steels with a specified tensile strength of greater than 950 N/mm^2 are not to be used due to the risk of hydrogen embrittlement.

5.1.4 For steerable pod units, an integral slewing ring is to be arranged at the upper extremity of the strut to provide support for the slewing bearing.

5.1.5 The strut is to have a smooth transition from the upper mounting to the lower hydrodynamic sections.

5.1.6 For fabricated structures, vertical and horizontal plate diaphragms are to be arranged within the strut and, where necessary, secondary stiffening members are to be arranged.

5.1.7 Pod unit structure scantling requirements are shown in *Table 4.5.1 Podded propulsion unit - fabricated structure requirements*. Where the scantling requirements in *Table 4.5.1 Podded propulsion unit - fabricated structure requirements* cannot be satisfied, direct calculations carried out in accordance with *Vol 2, Pt 4, Ch 4, 5.3 Direct calculations* may be considered.

Table 4.5.1 Podded propulsion unit - fabricated structure requirements

Location	Requirement	Notes
Strut external shell plating	Thickness, in mm, is to be not less than:	The minimum thickness of plating diaphragms and primary webs within the strut is to be not less than the

	$t = 0,0063sf\,(h_7k)^{0,5}$	Rule requirement for the strut external plating. For internal diaphragms, panel stiffening is to be provided where the ratio of spacing to plate thickness (*s/t*) exceeds 100. Where there are no secondary members, *s* is to be replaced by *S*.
Strut primary framing	The section modulus in cm^3 is to be not less than: $z = 7,75h_7 l_e^2 Sk$	This does not apply to full breadth plate diaphragms.
Strut secondary stiffening	The section in cm^3 is to be not less than: $z = 0,0056h_7 l_e^2 sk$	This does not apply to full breadth plate diaphragms.
Cylindrical pod body external shell plating	Thickness, in mm, is to be not less than: $t = 3,0R\mathrm{g}\,(h_7k)^{0,5}$	
Symbols		
f = panel aspect ratio correction factor = $[1,1 - s/(2500S)]$ h_7 = $(T + C_w + 0.014V^2)$ k = local higher tensile steel factor, as in *Vol 1, Pt 6, Ch 2 Design Tools* l_e = effective span of the member under consideration, in metres s = the frame spacing of secondary members, in mm C_w = design wave amplitude, in metres, as in *Table 3.2.1 Ship motions* in Vol 1, Pt 5, Ch 3 R_g = mean radius of pod body tube, in metres S = the spacing of primary members, in metres T = the vessel scantling draft, in metres, as in *Vol 1, Pt 3, Ch 1, 5.2 Principal particulars* V = maximum design speed, in knots, as in ~~*Vol 1, Pt 3, Ch 3, 2.11 Rudder force, FR*~~ *Vol 1, Pt 3, Ch 3, 2.6 Rudder force*		

5.1.8 The connection between the strut and the pod body should generally be effected through large radiused fillets in cast pod units or curved plates in fabricated pod units.

5.1.9 The structural response under the most onerous combination of loads is not to exceed the normal operational requirements of the propulsion or steering system components.

5.1.10 For cast pod structures, the elongation of the material on a gauge length of $5.65\sqrt{S_o}$ is to be not less than 12 per cent where S_o is the actual cross sectional area of the test piece.

5.1.11 In castings, sudden changes of section or possible constriction to the flow of metal during casting are to be avoided. All fillets are to have adequate radii which should, in general, be not less than 75 mm.

5.1.12 Castings are to comply with the requirements of *Ch 4 Steel Castings* or *Ch 7 Iron Castings* of the Rules for Materials.

■ *Section 5*
Pipe connections

5.6 Socket weld joints

5.6.1 Socket weld joints may be used in Class III systems with carbon steel pipes of any outside diameter. Socket weld fittings are to be of forged steel and the material is to be compatible with the associated piping. In particular cases, socket weld joints may be permitted for piping systems of Class I and II having outside diameter not exceeding 88,9 mm. Such joints are not to be used where fatigue, severe erosion or crevice corrosion is expected to occur or where toxic or asphyxiating media are conveyed, other than for carbon dioxide fire-extinguishing distribution piping.

5.6.2 The thickness of the socket weld fittings is to meet the requirements of *Vol 2, Pt 7, Ch 1, 6.1 Wrought steel pipes and bends 6.1.3*, but is to be not less than 1,25 times the nominal thickness of the pipe or tube. The diametrical clearance between the outside diameter of the pipe and the bore of the fitting is not to exceed 0,8 mm, and a gap of approximately 1,5 mm is to be provided between the end of the pipe and the bottom of the socket.

5.6.3 The leg lengths of the fillet weld connecting the pipe to the socket weld fitting are to be such that the throat dimension of the weld is not less than the nominal thickness of the pipe or tube.

5.6.4 Socket weld joints may be used in carbon dioxide fire-extinguishing system distribution piping only as permitted by *Vol 2, Pt 7, Ch 1, 5.11 ~~Carbon dioxide (CO2)~~ Piping for gaseous fire-extinguishing ~~system piping~~ systems* .

5.11 ~~Carbon dioxide (CO2)~~ Piping for gaseous fire-extinguishing ~~system piping~~ systems

5.11.1 The requirements of *Vol 2, Pt 7, Ch 1, 5.11 Piping for gaseous fire-extinguishing systems* are applicable to carbon dioxide (CO2) fire-extinguishing system piping arrangements. Piping arrangements for gaseous fixed fire suppression systems using gases other than carbon dioxide are to be agreed with LR.

~~5.11.1~~ 5.11.2 The piping for carbon dioxide fire-extinguishing systems is to comply with the requirements of *Chapter 5 - Fixed Gas Fire-Extinguishing Systems* of the FSS Code, as applicable. For purposes of classification, any use of the word 'Administration' in the FSS Code is to be taken to mean LR.

~~5.11.2~~ 5.11.3 Materials for the distribution manifolds between the carbon dioxide storage bottles and the discharge valves to each section and associated pipes, valves and fittings of high pressure systems are to be manufactured and tested in accordance with the requirements for Class I piping systems. Pipes are to meet the minimum wall thickness requirements of *Table 1.6.4 Minimum thickness for steel pipes* and the manifold system is to be hydraulically tested to a pressure of 190 bar. A high pressure system is defined as a system where the carbon dioxide is stored at ambient temperature. Materials for the distribution manifolds between the carbon dioxide storage vessel(s) and the discharge valves to each section and associated pipes, valves and fittings of low pressure systems are to be manufactured and tested in accordance with the requirements for Class II piping systems and the manifold system is to be hydraulically tested to a pressure of 33 bar. A low pressure system is defined as a system where the carbon dioxide is stored at a working pressure in the range of 1,8 N/mm^2 to 2,2 N/mm^2.

~~5.11.3~~ 5.11.4 Piping downstream of the distribution valve(s) for high pressure systems is to be manufactured and tested in accordance with the requirements for Class II piping and is to meet the minimum wall thickness requirements of *Table 1.6.4 Minimum thickness for steel pipes*. After installation the distribution system is to be leak tested at a pressure of 6 bar. Piping downstream of the distribution valve(s) for low pressure systems is to be manufactured and tested in accordance with the requirements for Class III piping. After installation the distribution system is to be leak tested at a pressure of 6 bar. Class III piping may be used for open-ended distribution piping downstream of the distribution valve(s) of high pressure systems where agreed by LR and where meeting the minimum wall thickness requirements of Table 1.6.4 and where a minimum of ten per cent of the piping is hydraulically tested at a pressure of 125 bar. This testing is to be carried out before installation.

~~5.11.4~~ 5.11.5 Any part of the carbon dioxide fire-extinguishing system piping is to be of galvanised steel or of corrosion-resistant steel. Where full penetration butt welding is used, the pipe is to be protected against corrosion in the area of the weld seam after welding. The process for protecting the pipe internally against corrosion is to be of an approved type. All pipes are to be arranged to be self-draining. Where pipes are to be led into refrigerated spaces, this is subject to special consideration. The ends of distribution pipes downstream of the distribution valve(s) are to extend at least 50 mm beyond the last nozzle and are to be fitted with a dirt trap consisting of an open-ended tee with a capped nipple.

~~5.11.5~~ 5.11.6 If it is necessary for carbon dioxide pipes to pass through accommodation spaces, the pipe is to be seamless and is to meet the requirements for Class II pipes. Joints are to be made only by welding and the pipes are to be hydraulically tested after installation at a pressure of 50 bar.

~~5.11.6~~ 5.11.7 The following means are permitted for making joints on carbon dioxide fire-extinguishing system piping;

(a) Full penetration butt welding, where the pipe is galvanised, *see Vol 2, Pt 7, Ch 1, 5.11 ~~Carbon dioxide (CO2)~~ Piping for gaseous fire-extinguishing ~~system piping 5.11.4~~ systems 5.11.5* .

(b) Couplings as permitted by *~~Vol 2, Pt 7, Ch 1, 5.10 Mechanical connections for piping 5.10.1~~ Table 1.5.3 Application of mechanical joints*.

(c) Cone connections.

(d) Tapered screw joints, where allowed by *Vol 2, Pt 7, Ch 1, 5.11 ~~Carbon dioxide (CO2)~~ Piping for gaseous fire-extinguishing ~~system piping 5.11.9~~ systems 5.11.10* and where meeting the requirements of *Vol 2, Pt 7, Ch 1, 5.11 ~~Carbon dioxide (CO2)~~ Piping for gaseous fire-extinguishing ~~system piping 5.11.9~~ systems 5.11.10* .

(e) Flanged joints.

(f) Socket weld joints to acceptable National Standards and where allowed by *Vol 2, Pt 7, Ch 1, 5.11 ~~Carbon dioxide (CO2)~~ Piping for gaseous fire-extinguishing ~~system piping 5.11.7~~ systems 5.11.8* and where meeting the requirements of *Vol 2, Pt 7, Ch 1, 5.11 ~~Carbon dioxide (CO2)~~ Piping for gaseous fire-extinguishing ~~system piping 5.11.8~~ systems 5.11.9* .

~~5.11.7~~ 5.11.8 Socket weld joints of an approved type may be used downstream of the distribution valve(s), provided that the requirements for materials and limitations on outside diameter applicable for Class II piping are applied.

~~5.11.8~~ 5.11.9 Where socket weld joints are utilised, the pipes in the way of the weld joints are to be adequately supported and the joints are to be located where they are visible. Where welding is to be carried out in situ, the piping is to be kept clear of adjacent structures to allow sufficient access for preheating and welding, which is to be carried out in accordance with approved procedures.

~~5.11.9~~ 5.11.10 Threaded joints are only allowed inside protected spaces and in carbon dioxide bottles storage rooms. They should have no exposed screw threads and any thread sealing medium should be selected as to ensure no protrusions or debris might be produced into the pipe.

Volume 2, Part 7, Chapter 3
Machinery Piping Systems

■ *Section 5*
Steam piping systems

5.2 Drainage

5.2.1 The slope of the pipes and the number and position of the drain valves or cocks are to be such that water can be efficiently drained from any portion of the steam piping system when the ship is in normal trim and is either upright or has a list of up to 5°.

5.2.2 Arrangements are to be made for ready access to the drain valves or cocks.

5.2.3 For the drainage of boiler and exhaust gas economiser safety valves, *see Vol 2, Pt 8, Ch 1, 15.2 Safety valves 15.2.8* and *Vol 2, Pt 8, Ch 1, 15.2 Safety valves ~~15.2.9~~* .

Volume 2, Part 8, Chapter 1
Steam Raising Plant and Associated Pressure Vessels

■ *Section 15*
Mountings and fittings for cylindrical and vertical boilers, steam generators, pressurised thermal liquid and pressurised hot water heaters

15.2 Safety valves

15.2.1 Boilers and steam generators are to be fitted with not less than two safety valves, each having a minimum internal diameter of 25 mm, but those having a total heating surface of less than 50 m^2 may have one valve not less than 50 mm diameter. Small oil fired package boilers of the once through coil type used for auxiliary or domestic purposes, where the feed pump capacity limits the output, may have one safety valve not less than 19 mm internal diameter, or two safety valves with internal diameters not less than 16 mm, provided the capacity is in accordance with *Vol 2, Pt 8, Ch 1, 15.2 Safety valves ~~15.2.13~~ 15.2.11*

15.2.2 The valves, spindles, springs and compression screws are to be so encased and locked or sealed that the safety valves and pilot valves, after setting to the working pressure, cannot be tampered with or overloaded in service.

15.2.3 Valves are to be so designed that in the event of fracture of springs they cannot lift out of their seats.

15.2.4 Easing gear is to be provided for lifting the safety valves and is to be operable by mechanical means at a safe position from the boiler or engine room platforms.

15.2.5 Safety valves are to be made with working parts having adequate clearances to ensure complete freedom of movement.

15.2.6 Valve seats are to be effectively secured in position. Any adjusting devices which control discharge capacity are to be positively secured so that the adjustment will not be affected when the safety valves are dismantled at surveys.

15.2.7 All the safety valves of each boiler and steam generator may be fitted in one chest, which is to be separate from any other valve chest and is to be connected directly to the shell by a strong and stiff neck, the passage through which is to be of cross-sectional area not less than the aggregate area of the safety valves in the chest in the case of full lift valves, and one-half of that area in the case of other valves. For the meaning of aggregate area, *see Vol 2, Pt 8, Ch 1, 15.2 Safety valves ~~15.2.13~~ 15.2.11* .

15.2.8 For each safety valve, an individual unrestricted drain is to be provided. The drain pipe is to be fitted to the lowest part of the discharge side of the safety valve; it is to be below the level of the valve seat and is to be independently led with a continuous fall to a place where the high temperature steam and/or condensate can discharge, visibly clear of the boilers, and where it cannot cause injury. No valves or cocks are to be fitted to these drain pipes. The bore of the drain pipes is not to be less than 19 mm. Where a drain pipe of 19 mm is impracticable, smaller drain pipes may be considered.

~~15.2.9 Safety valves for shell type exhaust gas steaming economisers are to incorporate fail safe features which will ensure operation of the valve even with solid matter deposits on the valve and guide, or features that will prevent the accumulation of solid matter in way of the valve and in the clearance between the valve spindle and guide. Alternatively, if the fitted valves do not incorporate the features described then a bursting disc discharging to a suitable waste steam pipe is to be fitted in addition to the valves. These bursting discs are to function at a pressure not exceeding 1,25 times the economiser approved design pressure and are to have sufficient capacity to prevent damage to the economiser when operating at its design heat input level.~~

~~15.2.10~~ 15.2.9 To avoid the accumulation of ~~solid matter deposits~~ condensate on the outlet side of the safety valves and bursting discs required by *Vol 2, Pt 8, Ch 1, 15.2 Safety valves ~~15.2.9~~* , the discharge pipes ~~and~~ and/ or safety ~~valve/bursting disc~~ valve housings are to be fitted with drainage arrangements from the lowest part, directed with continuous fall to a position clear of the economiser where it will not pose a threat to either personnel or machinery. No valves or cocks are to be fitted in the drainage arrangements. The drainage arrangements required by *Vol 2, Pt 8, Ch 1, 15.2 Safety valves 15.2.8*may be accepted as meeting these requirements where the arrangements comply with this paragraph.

~~15.2.11~~ 15.2.10 Full particulars of the proposed arrangements are to be submitted for consideration.

~~15.2.12 Where the receiver is fitted with safety valves to relieve the steam output of the economiser and the economiser cannot be isolated from the receiver the requirements of *Vol 2, Pt 8, Ch 1, 15.2 Safety valves 15.2.9* may be waived.~~

~~15.2.13~~ 15.2.11 The designed discharge capacities of the safety valves on each boiler and steam generator are to be found from the following formulae:

(a) Saturated steam safety valves:

$$E = \frac{AC(p + 1,03)}{98,1}$$

Superheated steam safety valves:

$$E = \frac{AC(p + 1,03)}{98,1}\sqrt{\frac{V_s}{V_H}}$$

where

p = set pressure, in bar gauge

A = for ordinary, high lift or improved high lift safety valves, the aggregate area, in mm^2, of the orifices through the seatings of the valves, neglecting the area of guides and other obstructions

= for full lift safety valves, the net aggregate area, in mm^2, through the seats after deducting the area of the guides or other obstructions when the valves are fully lifted

C = 4,8 for valves of ordinary type having a minimum lift of $\frac{D}{24}$

= 7,2 for valves of high lift type, having a minimum lift of $\frac{D}{16}$

= 9,6 for valves of improved high lift type having a minimum lift of $\frac{D}{12}$

= 19,2 for valves of full lift type having a minimum lift of $\frac{D}{4}$

D = bore of valve seat, in mm

E = the maker's specified peak load evaporation, in kg/hour (including all evaporation from water walls, integral, or steaming economisers and other heating surfaces in direct communication with the boiler)

V_H = specific volume of superheated steam (m^3/kg)

V_S = specific volume of saturated steam (m^3/kg).

~~15.2.14~~ 15.2.12 When the discharge capacity of a safety valve of approved design has been established by type tests, carried out in the presence of the Surveyors or by an independent authority recognised by LR, on valves representative of the range of sizes and pressures intended for marine application, consideration will be given to the use of a constant higher than C = 19,2, based on 90 per cent of the measured capacity up to a maximum of C = 45 for full lift safety valves.

~~15.2.15~~ 15.2.13 Pressurised thermal liquid and pressurised hot water heaters are to be provided with a safety relief device. The safety valve is to be designed and constructed in accordance with a relevant National or International Standard acceptable to LR.

15.3 Waste steam pipes

15.3.1 For ordinary, high lift and improved high lift type valves, the cross-sectional area of the waste steam pipe and passages leading to it is to be at least 10 per cent greater than the aggregate area of the safety valves as used in the formulae in *Vol 2, Pt 8, Ch 1, 15.2 Safety valves ~~15.2.13~~ 15.2.11* . For full lift and other approved valves of high discharge capacity, the cross-sectional area of the waste steam pipe and passages is to be not less than 0,1C times the aggregate valve area.

15.3.2 The cross-sectional area of the main waste steam pipe is to be not less than the combined cross-sectional areas of the branch waste steam pipes leading thereto from the boiler safety valves.

15.3.3 Waste steam pipes are to be led to the atmosphere and are to be adequately supported and provided with suitable expansion joints, bends or other means to relieve the safety valve chests of undue loading.

15.3.4 The scantlings of waste steam pipes and silencers are to be suitable for the maximum pressure to which the pipes may be subjected in service, and in any case not less than 10 bar.

15.3.5 Silencers fitted to waste steam pipes are to be so designed that the clear area through the baffle plates is not less than that required for the pipes.

15.3.6 The safety valves of each exhaust gas heated economiser and exhaust gas heated boiler which may be used as an economiser are to be provided with entirely separate waste steam pipes.

15.3.7 External drains and exhaust steam vents to atmosphere are not to be led to waste steam pipes.

15.3.8 It is recommended that a scale trap and means for cleaning be provided at the base of each waste steam pipe.

15.12 Additional requirements for ~~shell type~~ shell-type exhaust gas ~~steaming~~ heated economisers

15.12.1 The requirements of *Vol 2, Pt 8, Ch 1, 15.12 Additional requirements for shell-type exhaust gas heated economisers* are applicable to shell-type exhaust gas heated economisers that are intended to be operated in a flooded condition and that may be isolated from the steam plant system.

~~15.12.1~~ 15.12.2 The design and construction of shell type economisers are to pay particular attention to the welding, heat treatment and inspection arrangements at the tube plate connection to the shell.

~~15.12.2~~ 15.12.3 Every shell type economiser is to be provided with removable lagging at the circumference of the tube end plates to enable ultrasonic examination of the tube plate to shell connection.

15.12.4 Every shell-type exhaust gas heated economiser is to be provided with a means of indicating the internal pressure. A means of indicating the internal pressure is to be located so that the pressure can be easily read from any position from which the pressure may be controlled.

~~15.12.3~~ 15.12.5 Every economiser is to be provided with arrangements for pre-heating and de-aeration, and addition of water treatment or combination thereof, to control the quality of feed water to within the manufacturer's recommendations.

~~15.12.4~~ 15.12.6 The manufacturer is to provide operating instructions for each economiser which is to include reference to:

- Feed water treatment and sampling arrangements.
- Operating temperatures-exhaust gas and feed water temperatures.
- Operating pressure.
- Inspection and cleaning procedures.
- Records of maintenance and inspection.
- The need to maintain adequate water flow through the economiser under all operating conditions.
- Periodical operational checks of the safety devices to be carried out by the operating personnel and to be documented accordingly.
- Procedures for using the exhaust gas economiser in the dry condition.
- Procedures for monitoring, cleaning, maintenance and overhaul of safety valves.
- Emergency operating procedures.

■ *Section 2*
Hull strengthening requirements

2.13 Rudder and steering arrangements

2.13.1 Rudder posts, rudder horns, solepieces, rudder stocks and pintles are to be dimensioned in accordance with *Vol 1, Pt 3, Ch 3, 2 Rudders* and ~~*Vol 1, Pt 3, Ch 3, 2.13 Rudder bending moment, MR* as appropriate. Rudder scantlings are to be determined in accordance with~~ *Vol 1, Pt 3, Ch 3, 2.9 Rudder strength calculation* as appropriate. Rudder scantlings are to be determined in accordance with *Vol 1, Pt 3, Ch 3, 2 Rudders* using the basic stock diameter, δ_S. The speed used in the calculations is to be the service speed or that given in *Table 1.2.8 Minimum speeds*, whichever is the greater. When used in association with the speed given in *Table 1.2.8 Minimum speeds*, the hull form factor and rudder profile coefficients are to be taken as 1,0.

Table 1.2.8 Minimum speeds

Ice Class	Minimum speed, in knots
1AS	20
1A	18
1B	16
1C	14

2.13.2 For double plate rudders, the minimum thickness of plating and horizontal and vertical webs in the main ice belt zone is to be determined as for shell plating in the aft region in accordance with *Vol 3, Pt 1, Ch 1, 2.4 Shell plating 2.4.1*. For the horizontal and vertical webs the corrosion-abrasion increment, *c*, need not be added.

2.13.3 For ice classes **1AS** and **1A**, the rudder head and the upper edge of the rudder are to be protected against ice pressure by an ice knife, or equivalent.

2.13.4 Due regard is to be paid to the method of securing the rudder in the centreline position when backing into ice. Where possible, rudder stoppers working on the blade or rudder head are to be fitted.

2.13.5 Where an ice class notation is included in the class of a ship, the nozzle construction requirements as defined in *Vol 1, Pt 3, Ch 3, 5 Fixed and steering nozzles, bow and stern thrust units, ducted propellers*are to be upgraded to include abrasion allowance as follows:

Ice Class	Thickness increment
1AS	5 mm
1A	4 mm
1B	3 mm
1C	2 mm

However, the thickness of the shroud plating is not to be less than the shell plating for the aft region as obtained from *Vol 3, Pt 1, Ch 1, 2.4 Shell plating 2.4.1* taking frame spacing *s* in the formula as 500 mm.

2.13.6 The scantlings of the stock, pintles, gudgeon and solepiece associated with the nozzle are to be increased on the basis given in *Vol 3, Pt 1, Ch 1, 2.13 Rudder and steering arrangements 2.13.1*. However, the diameter of the nozzle stock is to be not less than that calculated in the astern condition taking the astern speed as half the speed given in *Table 1.2.8 Minimum speeds* or the actual astern speed, whichever is the greater.

2.13.7 Nozzles with articulated flaps will be subject to special consideration.

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